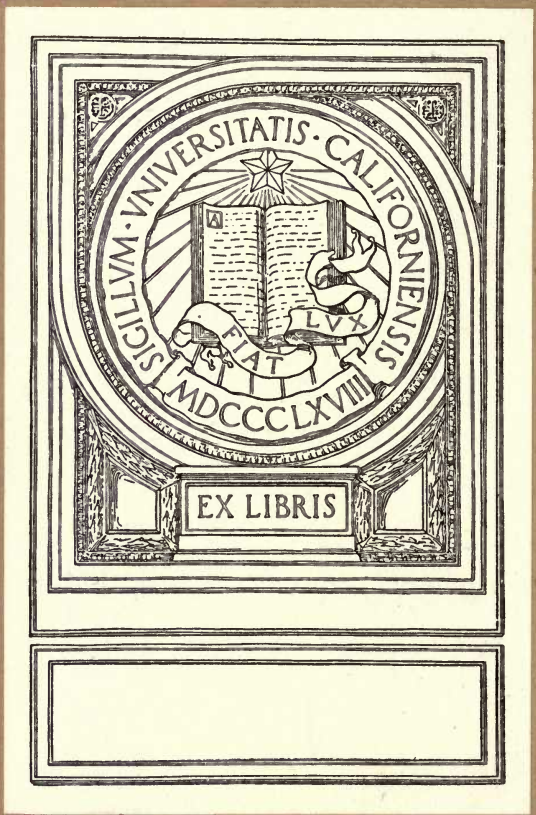
UC-NRLF
$B 315 432

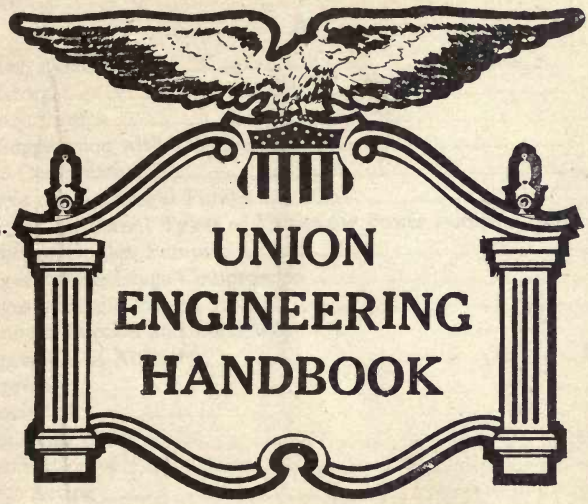
UNION ENGINEERING HANDBOOK

PUMPING MACHINERY
AIR COMPRESSORS
CONDENSERS

UNION STEAM PUMP CO.
BATTLE CREEK, MICHIGAN
U. S. A.


SIGILLVM · VNIVERSITATIS · CALIFORNIENSIS
FIAT
LVX
MDCCCLXVIII
EX LIBRIS

Paul Christiansen.



UNION ENGINEERING HANDBOOK

PUMPING MACHINERY
AIR COMPRESSORS
CONDENSERS

COMPILED BY

E. P. ORDWAY, M. E.

SIXTH EDITION

PRICE $5.00

UNION STEAM PUMP CO.

BATTLE CREEK, MICHIGAN

U. S. A.

[illegible]

PRINTED IN THE U. S. A.

TJ900
07
1921

COPYRIGHT, 1921,
BY THE
UNION STEAM PUMP COMPANY
BATTLE CREEK, MICHIGAN
U. S. A.

ELLIS PUBLISHING CO.
PRINTERS, ENGRAVERS
ELECTROTYPERS, BINDERS
AND RULERS
BATTLE CREEK, MICHIGAN

INDEX

A

Accessories, Electrical 252–255
Accumulators 341
Acid Sludge Pumps 474
Actual Compression with Clearance 8
Adiabatic Compression 4
Advantages of Centrifugal Pumps 97
Advantages of Different Types of Drives for Power Pumps 382
Advantages of Duplex Pumps 266
Advantages of Multistage Compression 12
Advantages of Single Pumps 263
Air Chambers, Suction and Discharge 290
Air Compression at Altitudes 17 70, 73
Air Compressors
- Capacity of 18
- Circulating Water for 26
- Classification of 36
- Direct Acting 345, 375
- Displacement of 18
- Drives 36
- Duplex, Belt and Steam 41–45
- Effect of Intake Temperatures 70
- Efficiency, Volumetric and Mechanical 14, 17, 19
- Foundations for 21
- Gas or Gasoline Extraction 39B
- General Construction 26
- Inspection and Cleaning of 25
- Installation and Operation of 20
- Location of 20
- Lubrication 15, 22
- Single, Belt and Steam 39–40
- Steam Consumption 37
- Uses 47–51
- Valve Gear 32
- Vertical 45–46

Air Compressors, Gas or Gasoline Extraction 39B
Air Consumption of Tools and Machines 87–88
Air and Circulating Pumps 213
Air Cylinders 32
Air Cylinders, Displacement of 65
Air Drills 78
Air, Flow Through Round Holes 77
Air Inlet Piping 21
Air Lift Advantages 57
Air Lift Calculations 61–62
Air Lift Data Table 63
Air Lift Installation 59
Air Lift Piping 60
Air Lift Terms 57
Air, Loss of Pressure in Pipes and Valves 80–83
Air Pipes, Carrying Capacity 79

M73420

Air Receivers 22, 46
Air, Removing Moisture 14
Air Valves 30–31
Air, Volumes of Free Air 84
Air, Weight at Various Pressures 85
Alternating Current 233
Alternating Current Motors 234
Amperes 230
Ampere Ratings A. C. Motors 238
Ampere Ratings D. C. Motors 239
Areas of Circles 427–431
Areas, Ratio of 307–308
Asphaltic Base Oils 24–453
Automatic Centrifugal Pumps and Receivers, Motor Driven 107A
Automatic Feed Pumps and Receivers, Burnham 350
Automatic Feed Pumps and Receivers, Union Duplex 351B

B

Ball Valves 311
Barometer, Reduction to Sea Level 332
Baume Scales 447
Belt Driven Dry Vacuum Pumps 215, 390
Better Lubrication 15
Bevel Seat Valves 311
Boiler Feed Pumps 314
Boiler Feed Pumps, Centrifugal 106
Boiler Feed Pumps, Duplex 350–351
Boiler Feed Pumps, Simplex 348–349
Boiler Feed Pump and Receivers, Centrifugal 107A
Boiler Feed Pump and Receivers, Duplex 351B
Boiler Feed Pump and Receivers, Simplex 350
Boiler Feed Pump Ratings, Simplex and Duplex 315
Boiling Points of Liquids 331
Boyle's Law 3
Brake H. P. of Air Compressors 38
Brass Tubing, Weight of 218–225
Bronze Valves 310
Burnham Steam Pumps 260
Burnham Valve Gear, Directions for Setting 262

C

Cables and Ropes 442
Calorific Power of Fuels 437
Capacity of Air Compressors 18
Capacity of Cylinders 306
Capacity of Pumps 294
Capacity of Pumps, Table 305
Cargo Oil Pumps, power driven 393B
Cargo Oil Pumps, steam driven 351B
Casing Head Gasoline 456
Center Packed Plunger Pumps 271, 352
Centrifugal Pumps
  Advantages 97
  Characteristics and Explanations 117–120

Centrifugal Pumps—*Continued*
Data 100
Directions for Installing and Operating 129–130
Efficiency 121
Head Determination 108–109
Horse Power Calculations 132
Ifs 131
Multi-Stage 107
Power Consumption 141
Priming 126–127
Pumps and Receivers 107A
Relation of Capacity, Head, Speed and Horse Power 121–122
Side Suction Pumps 103–104
Sump Pumps, Single and Duplex 107B
Uses 97–99
Centrifugal House Pump 107C
Centrifugal Pumps, Double Suction, High Speed 106
Centrifugal Pumps, Double Suction, Horizontally Split Case 105
Centrifugal Pumps, Multistage 107
Centrifugal Pumps for Paper Stock 107D
Centrifugal Pumps and Receivers 107A
Centrifugal Pumps 90
Characteristics of Average Oils 476
Characteristics of Tested Oils 475
Charles' Law 3
Chemical Properties of Petroleum 451
Choice of Condensers 181–182
Circles, Areas and Circumference 427–431
Circular Measure 441
Circulating Water for Compressors 26
Circumference and Areas of Circles 427–431
Clapper Valves 311
Classification of Air Compressors 36
Co-Efficients of Linear Expansion 432
Comparative Economy of Turbines and Engines 179
Comparative Table U. S. and Metric System 435–436
Comparison of Hydrometer Scales 333–334
Comparison of Thermometers 424
Compound Pumps, Center Packed 357
Compound Pumps, End Packed 358
Compound Pumps, Heavy Piston 355
Compound Pumps, Light Service 356
Compound Pumps, Pot Valve 359
Compound Steam Cylinder Calculations 280-281
Compound Steam Cylinder Pumps 279
Compound Steam Cylinder Table 282

Compressed Air
Data 52
Multipliers for 71
Principles of 2
Table for Hoisting Engines 74
Table for Motors 75
Table for Pumps 76
Temperature and Mean Pressure 69

Compression— *Continued*
Adiabatic 4–5
Cylinder Ratio, Two-Stage Compression 16
Efficiency 19
Isothermal 6–7
Loss of Work Due to Heat 68
Multi-Stage 12
Single and Two Stage 10, 66
Temperature of Cylinders 23
Three and Four Stage 67
With Clearance 8
Compressor Installation and Operation 20
Condensers
Auxiliaries 186, 187, 208–215
Cooling Water 185–205
Installation 227–228
Jet 188–190
Power Saving 167–173
Principles 166, 183–184
Selection of 181–182
Surface 197–201
Surface, High-Vacuum Type 203
Connecting Rods 28
Consumption of Electricity for Pumping 141
Consumption of Gasoline for Pumping 141
Contents of Cylinders 79
Contents of Cylindrical Tanks 416
Convenient Equivalents 156–158
Cooling Water for Condensers 185
Cooling Water for Surface Condensers 205
Correct Cylinder Ratios, Two Stage Compression 16
Cost of Compressed Air, Electric Driven Compressors 55
Cost of Compressed Air, Gasoline Driven Compressors 55
Cost of Compressed Air, Steam Driven Compressors 54
Cost of Pumping 133
Cracking Process in Oil Refining 455
Crank and Fly Wheel Pumps, Classification 383
Crankshaft 27
Creating a Vacuum in a Closed Tank 404–405
Cross Cracking Unit, flow diagram 457
Crossheads 29
Cubic Feet of Air for Drills 78
Cubic Measure 441
Cylinders
Capacity of, Gallons 306
Contents of, Cubic Feet 79
Piston Displacements, Cubic Feet 86
Cylinder Temperatures 23

D

Data for Centrifugal Pumps 100
Data for Deep Well Pumps 347
Data for Direct Acting Pumps 346
Data for Power Pumps and Crank and Fly Wheel Pumps 386
Decimal Equivalents 425

Decimal Equivalents of Millimeters 426
Deep Well Pumps 342
Deep Well Pumps Data 347
Deep Well Pump Table 377
Density of Gases and Vapors 70
Determination of Total Head of Centrifugals 108-109
Diagram of Air Lift 58
Diagrammatic Arrangement of Triple Effect Evaporator 324
Diameters of Pulleys 154
Direct Acting Air Compressors 345
Direct Acting Air Compressor Table 375
Direct-Acting Pumps Data 346
Direct Acting Steam Pump Classification 258
Direct Current 234
Direct Current Motors 238
Directions for Installing Centrifugal Pumps 129
Directions for Operating Centrifugal Pumps 130
Directions for Setting the Burnham Valve Gear 262
Directions for Setting Valves of Duplex Pumps 266
Discharge Nozzles 115
Discharge Piping 22
Discharge of Water from Nozzles 149
Displacement of Air Compressors 18
Displacement of Air Cylinders 65
Displacement of Pumps 293
Displacement of Pumps for Evaporator Work 322
Ditches 159
Distillation of Petroleum 454
Double Suction Pumps 95
Double Suction Pumps, Horizontally Split Case Type 105-106
Drier Air 14
Dry Measure 441
Dry Vacuum Pumps, Duplex Belt 394
Dry Vacuum Pumps, Duplex Steam 402
Dry Vacuum Pumps, Single Belt 390
Dry Vacuum Pumps, Single Steam 399
Duplex Air Compressors:
Class DBL Belt 41
Class BD Belt 41
Class DSL Steam 42
Class SD Steam 42
Class DBTL 2-Stage, Belt 43
Class BDT 2-Stage, Belt 43
Class DSTL 2-Stage, Steam 44
Class SDT 2-Stage, Steam 44
Class CTD Cross Compound, 2-Stage 45
High Pressure Gas Compressors 39B
Duplex Boiler Feed or Pressure Pumps 351
Duplex Pressure Oil Pumps 351A
Duplex Pumps, Directions for Setting Valves 266
Duplex Crank and Fly Wheel Dry Vacuum Pumps 402
Duplex Crank and Fly Wheel Wet Vacuum Pumps 401
Duplex Dry Vacuum Pumps, Belted 394

Duplex Light Service Pumps 363
Duplex Power Oil Line Pumps 393A
Duplex Power Pumps 393
Duplex Piston Pumps, Horizontal 264
Duty 300–302
Duty Table 303

E

Economy Rating of Evaporators 329
Economy of 28" Vacuum over 26" 176
Effect of Intake Temperatures 70
Efficiency of Air Compression at Altitudes 70
Efficiency of Centrifugal Pumps 121
Efficiency of Induction Motors 237
Efficiency of Power Pumps 384
Elbows, Loss of Pressure 83
Electrical Accessories 252–255
Electrical Control, Manual and Automatic for
Air Compressors 240–252
Centrifugal Pumps 240–252
Power Pressure Pumps 240–252
Power Vacuum Pumps 240–252
Electrical Equivalents 233
Electrical Units 230–232
Electric Gear Driven Pumps 382
Elevations of Various Cities 332
Elevator Pumps 318
End Packed Plunger Pumps 272
End Packed Plunger Pump Table 353
End Packed Pot Valve Plunger Pump Table Simplex 354
End Packed Pot Valve Plunger Pump Table Duplex 354
End Packed Pot Valve Pumps 273
Engines, Horsepower of 417
Engines, Steam Consumption 174
Equivalents, Table 156–158
Evaporation 411
Evaporation in a Vacuum 321
Evaporator, Diagrammatic Arrangement of Triple Effect 324
Evaporators, Multiple Effect 323
Evaporators, Rating of 329
Evaporator Work, Displacement of Pumps 322

F

Factors of Evaporation 411
Feed Pumps, Ratings, Simplex and Duplex 315
Feed Pumps and Receivers 316
Feed Pumps and Receivers Table 350
Feed Water Pumps 314
Feed Water Pumps, Duplex 350–351
Feed Water Pumps, Simplex 348–349
Flanges 412–413
Flow Diagram Cross Cracking Coil 457
Flow of Air Through Round Holes 77
Flow of Water in Flumes 164

Foamite Pump 473
Foundations for Air Compressors 21
Friction of Oil in Pipes, Formula 475
Friction of Oil in Pipes, Table 477–478
Friction of Paper Stocks in Pipes 443–446
Friction of Pipe Fittings 148
Friction of Water in Elbows 146–147
Friction of Water in Pipes 144–145
Fuel, Calorific Power of 437
Full Load Speed of Induction Motors 237

G

Gain in Compounding Pumps 280
Gain in Thermal Efficiency 172
Gas or Gasoline Extraction Compressors 39B
Gases and Vapors, Density of 70
Gasoline Pumps 464
Gear Driven Pumps 382
Gears and Pulleys, Rules for Sizes 153
General Construction of Air Compressors 26
Gunter's Chain 442

H

Handling Hot Water 287
Head in Feet of Water and Mercury Equivalents 142–143
Head, Measurement of 292
Heating Systems, Steam Required 336
High Vacuum Pumps 320
High Vacuum Pump Table 368–370
High Vacuum Surface Condensers 203
Horizontal Duplex Boiler Feed or Pressure Pumps 351
Horizontal Mine Pumps 365
Horizontal Piston Pumps, Duplex 269
Horizontal Piston Pumps, Simplex 267
Horsepower of Centrifugal Pumps, Calculation 132
Horsepower, Efficiency and Temperatures of Single and Two Stage Compression 66
Horsepower, Efficiency and Temperatures of Three and Four Stage Compression 67
Horsepower of Engines 417
Horsepower—Hour 232
Horsepower of Motors 233
Horsepower of Power Pressure Pumps, Calculation of 384–385
Horsepower of Power Vacuum Pumps, Calculation of 385
Horse Power Ratings, Duplex Boiler Feed Pumps 315
Horse Power Ratings, Simplex Boiler Feed Pumps 315
Hot Oil Pumps
- Forged Steel Compound 468
- Forged Steel Duplex 469
- Forged Steel Simplex 468
- Twin Forged Steel Compound Plunger 472
- Twin Forged Steel Piston 470
- Twin Forged Steel Plunger 471
- Twin Valve Pot 466–467
- Valve Pot Duplex 465
- Valve Pot Simplex 465

House Pump, Centrifugal 107C
Hydrant and Hose Stream Data 150–151
Hydraulic Efficiency 295
Hydraulic Pressure Pumps 274, 341
Hydraulic Pressure Pumps, Cast Iron Cylinder Table 373
Hydraulic Pressure Pumps, Forged Steel Cylinder Table 374
Hydrometer Scales 333–334
Hyperbolic Logarithms 226

I

Impellers 91
Impeller Diagram 101
Impeller Theory 102
Indicated H. P. of an Air Compressor 38
Indirect Radiation 336
Induction Motors 234
Information for Air Lift 64
Inspection and Cleaning of Air Compressors 25
Intercooler 13
Inverted Suction Valve Vacuum Pumps 371
Irrigation 158
Irrigation Table 161–162
Isothermal Compression 6

J

Jet Condensers 188–190
Jet Condenser, Calculations 194–197
Jet Condenser Installation 191
Jet Condensers, Table 214
Joule 232
Joules' Law 3

K

Kilovolt 230
Kilovolt-Ampere 232
Kilowatt 231
Kilowatt-Hour 232

L

Land Measure 442
Light Service Horizontal Piston Pumps 360–361
Light Service Vertical Piston Pumps 362
Location of Compressors 20
Logarithms 419–420
Logarithms, Hyperbolic 226
Loss of Heat, Simple and Compound Compression 68
Loss of Pressure in Air Pipes, 75 lbs 80
Loss of Pressure in Air Pipes, 90 lbs 81
Loss of Pressure in Air Pipes, 100 lbs 82
Loss of Pressure in Air Valves, Tees and Elbows 83
Low Vacuum Pumps 335
Low Vacuum Pumps, Table 366–368
Lubrication
  Compressor Cylinder 15
  Steam Cylinder 25

M

Magma Pumps, Belt ... 391
Magma Pumps, Steam ... 344
Magma Pumps, Table ... 378
Materials and Manner of Fitting Centrifugal for Different Liquids ... 137–140
Materials and Manner of Fitting Reciprocating Pumps for Different Liquids ... 297–299
Measures of Length ... 442
Measurement of Power ... 116
Measurement of Speed ... 116
Measurement of Total Head ... 292
Measurement of Water ... 110
Mechanical Efficiency of Air Compressors ... 17
Mechanical Efficiency of Pumps ... 295
Melting Points ... 432
Mensuration of Surfaces and Volumes ... 433–434
Metal Compositions, Proportion of ... 438
Metallic Packing ... 313
Methods of Priming Centrifugal Pumps ... 126–127
Metric Conversion Table ... 436
Milk Pumps ... 343
Milk Pumps, Table ... 378
Millivolt ... 230
Mine Pumps ... 317, 364, 365
Most Economical Vacuum for Steam Engines ... 173
Most Economical Vacuum for Steam Turbines ... 174
Motors
- Alternating Current ... 233–238
- Direct Current ... 238–239
- Induction ... 236
- Selection of ... 240
- Slip Ring ... 235
- Squirrel Cage ... 235

Multiple-Effect Evaporators ... 323
Multipliers for Air Drills ... 78
Multipliers for Compressed Air ... 71
Multi-Stage Compression ... 12
Multi-Stage Turbine Pumps ... 107

N

Natural Trigonometric Functions ... 421–423
Nozzles for Water Measurement ... 115
Nozzles, Discharge of in G. P. M. ... 149

O

Ohm's Law ... 230
Oils, Characteristics of ... 475–476
Oils, Viscosity Chart ... 479
Oil
- For Air Compressors, Steam Cylinder ... 25
- For Air Cylinders ... 25
- Table ... 476
- Useful Information ... 475

Oil Pumps
Cargo Loading Pumps ... 474
Centrifugal Pumps for Light Oils ... 464
Centrifugal Gasoline Pumps ... 464
Duplex Pumps ... 461
Duplex Oil Line Pumps Steam ... 462
Duplex Oil Line Pumps Power ... 393A
Duplex Plunger High Pressure Pumps ... 463
Foamite Pumps ... 473
Forged Steel Hot Oil Compound ... 468
Forged Steel Hot Oil Duplex ... 469
Forged Steel Hot Oil Simplex ... 468
Multi-Stage for Oils and Water ... 464
Separate Chest ... 461
Simplex Piston ... 461
Simplex Plunger High Pressure ... 462
Sludge Pumps ... 474
Valve Pot Hot Oil Simplex ... 465
Valve Pot Hot Oil Duplex ... 465
Valve Pot Hot Oil Twin ... 466–467
Twin Forged Steel Hot Oil Piston ... 470
Twin Forged Steel Hot Oil Plunger ... 471
Twin Forged Steel Hot Oil Compound Plunger ... 472
Oils, Characteristics of ... 476
Open Pot Water Seal ... 313
Operation of Burnham Pumps ... 261
Operation of Turbines at High Vacuums ... 177–178
Outside Center Packed Pumps ... 271
Outside End Packed Pumps ... 272

P

Packing, Metallic ... 313
Paper Stock Pump, Centrifugal Type ... 107D
Paper Stock, Friction in Pipes ... 443–446
Paraffin Base Oils ... 24–453
Performance Factors of Pumps ... 292
Physical Properties of Petroleum ... 452
Pipe, Radiating Surface of ... 340
Piping, Dimensions and Weights ... 414–415
Piping, Discharge ... 22
Piston Speeds for Pumps ... 292
Pitot Tube ... 115
Plain Belted Power Pumps ... 381
Plunger Pumps ... 271, 272, 352–354
Pot Valve Pumps, Simplex ... 273, 354
Pot Valve Pumps, Duplex ... 274–354
Power Factor ... 232
Power Factor for Induction Motors ... 237
Power Magma Pumps ... 391
Power, Measurement of ... 116
Power Pumps, Advantages of Different Types of Drives ... 382
Power Pump Classification ... 380
Power, Pump Data ... 386
Power Pumps, Efficiency of ... 384

Power Pumps, Single Light Service 388
Power Pumps, Single Pressure 387
Power Pumps, Duplex 393
Power Pumps, Duplex Oil Line 393A
Power Pumps Single Enclosed 392
Power Required for Condenser Auxiliaries 212–213
Power Saving with Condenser 167–173
Pressure Conversion Factors 331
Pressure Loss in Air Pipes 80–83
Pressures for Stage Compression 15
Prime Movers for Centrifugals 123
Prime Movers, Steam Consumption 417
Priming Centrifugal Pumps 126–127
Principles of Compressed Air 2
Principles of Multiple Effect Evaporators 323–327
Principles of Surface Condensers 183–185
Properties of Saturated Steam for Condensers 206–207
Proportions of Various Compositions 438
Pulleys and Gears 153
Pulleys and Gears, Rules for Sizes 153
Pulleys, Diameter of 154
Pumping Cost 133
Pumping Liquids Other Than Water 134–136
Pumping With Compressed Air 57
Pumps for the Oil Industry 458
Pumps
- Advantages of Duplex 266
- Capacity of 305
- Centrifugal Pumps and Receivers 107A
- Displacement of 293
- Duplex Pumps and Receivers 351B
- Mechanical Efficiency 295
- Performance Factor 292
- Piston Speed 292
- Power 380
- Simplex Pumps and Receivers 350
- Steam 258
- Table of Capacity 305
- Valves 309
- Volumetric and Mechanical Efficiency 294–295

Q

Quantity of Lubricating Oil for Air Cylinders 25
Quantity of Lubricating Oil for Air Compressor Steam Cylinders 25

R

Radiating Surface of Pipe 340
Ratio of Areas 307–308
Ratio of Cylinders for Compound Pumps 280
Ratio of Submergence 59
Receivers, Air 22, 46
Reduction of Barometer to Sea Level 332
Refining of Oils 453

Relative Quantities of Water ... 152
Ring Packing for Fluid Pistons ... 312
Ropes and Cables ... 442
Rules for Size of Pulleys and Gears ... 153

S

Saturated Steam, Table ... 406–409
Saturated Steam at High Vacuum ... 176
Saturated Steam for Condenser Work ... 206–207
Selection of Motors ... 240
Separate Chest Pumps ... 351A–461
Short Belt Driven Power Pumps ... 381
Side Suction Volute Pumps ... 103–104
Simple Cylinder Pumps ... 278
Simple Cylinder Pumps, Calculation of ... 278
Single Belted Dry Vacuum Pumps ... 390
Single Crank and Fly Wheel Dry Vacuum Pumps ... 399
Single Crank and Fly Wheel Syrup Pumps ... 400
Single Crank and Fly Wheel Wet Vacuum Pumps ... 398
Single Enclosed Type Power Pumps ... 392
Single Piston Pattern Light Service Pumps ... 388
Single Piston Pattern Pressure Pumps ... 387
Single Pumps ... 259
Single Stage BL, Belt Driven Compressors ... 39
Single Stage SL, Steam Driven Compressors ... 39
Single Stage Compression at Altitudes ... 72
Single Suction Pumps ... 95
Sinking Pumps, Vertical ... 365
Size of Auxiliaries for Condensers ... 186–187
Size of Suction and Discharge Pipes ... 288
Slip of Induction Motors ... 236
Slip of Pumps ... 294
Slip Ring Motors ... 235
Sludge Pumps ... 474
Specific Gravity of Metals ... 438
Specific Gravity of Petroleum ... 452–476
Specific Heat at Constant Pressure ... 4
Specific Heat at Constant Volume ... 3
Speed of Induction Motors ... 236
Speed, Measurement of ... 116
Speed, Pumps ... 292
Squirrel Cage Motors ... 235
Standard Pattern Wet Vacuum Power Pumps ... 389
Steam Consumption of Air Compressors ... 37
Steam Consumption of Engines ... 174
Steam Consumption of Prime Movers ... 417
Steam Consumption of Turbines ... 178
Steam Cylinders ... 33
Steam Cylinder Lubrication ... 25
Steam Driven Dry Vacuum Pumps ... 215
Steam Economy of Pumps ... 304
Steam Indicated Horse Power ... 296
Steam Pumps ... 258
Steam Required by Heating Systems ... 336

Steam Table for Evaporator Work 330
Steam Turbines 125
Straight Distillation of Petroleum 454
Stuffing Boxes 313
Suction at Altitudes 287
Suction Air Chambers 291
Suction Defined 284
Suction Lift Diagram for Centrifugal Pumps 128
Suction Pipe 287
Sump Pumps 107B
Superheated Steam Table 410
Surface Condenser Calculations 185
Surface Condenser Construction 197
Surface to Condense Steam Under Different Conditions 204
Surface of Tubes for Condensers 216
Synchronous Motors 234
Syrup Pumps, Crank and Fly-Wheel 400

T

Table of Branch Pipes 79
Table of Degrees, Brix 448
Table Giving Injection Water, Vapor and Displacements for Jet Condensers 192–193
Table of Horse Power Ratings, Duplex Feed Pumps 315
Table of Horse Power Ratings, Simplex Boiler Feed Pumps 315
Tanks, Contents of 416
Tank Pumps 360–364
Tees, Loss of Pressure in 83
Tests of Turbines 178
Theoretical Capacity of Pumps 305
Theoretical Horse Power to Raise Water 155
Thermal Efficiency 172
Thermometers 424
Three Ring Fluid Piston Packing 312
Triple Effect Evaporator 328
Triplex High Pressure Milk Pumps 396
Triplex Plunger Pumps 395
Tubes, Surface in Square Feet 217
Turbine Pumps 92
Turbines, Operation of at Different Vacuum 177–178
Turbo-Generators, Water Rates 180
Twin Pumps 459
Two Stage Air Compression 10
Two Stage Compression at Altitudes 73
Two Stage Single Acting Belt Driven Compressors 40
Two Stage Double Acting Belt Driven Compressors 39A
Two Stage Single Acting Steam Driven Compressors 40
Two Stage Double Acting Steam Driven Compressors 39A
Types of Compound Pumps 283
Types of Drives for Air Compressors 36

U

U. S. Standard Flanges 412
U. S. Standard Flanges, Extra Heavy 413

U. S. System and Metric Comparative Tables 435–436
Useful Data 331
Useful Information, Oil 475
Uses of Centrifugal Pumps 97–99
Uses of Compressed Air 47
Use of Hydraulic Pressure Pumps 341

V

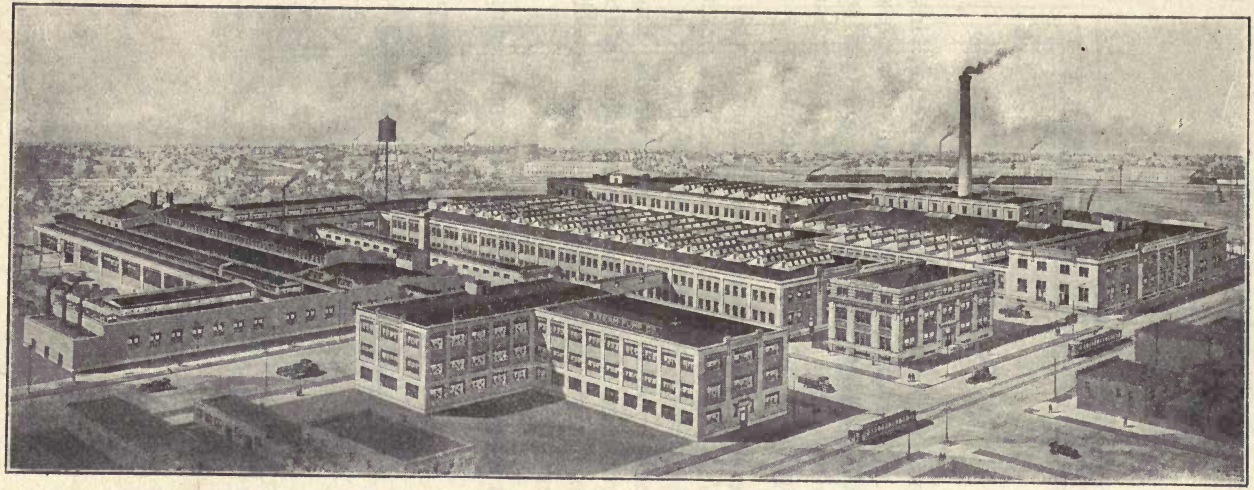
Vacuum at Sea Level 175
Vacuum, Economy of 28″ Vacuum Over 26″ 176
Vacuum, Feet Conversion Table 152
Vacuum for Steam Engines 173
Vacuum for Steam Turbines 174
Vacuum Heating Pumps, Calculation of 338
Vacuum Heating System 337
Vacuum in Closed Tank 404–405
Vacuum Pans 322
Vacuum Pumps
- Crank and Flywheel, Wet Vacuum 398–401
- Dry Vacuum 390, 394, 399, 402
- High-Vacuum, Uses 320
- Inverted Suction-Valve 371
- Low and High Classification 319–320
- Low-Vacuum, Uses 335
- Tables, Low and High 366–371
- Wet Vacuum, Power Driven 389
- Wet Vacuum, Steam Driven 366–372

Valves, Air 30–31
Valve Area 309
Valve Gear for Air Compressors 32
Valve Motion, Burnham Pumps 260
Valve Motion Duplex Pumps 264
Valves, Pump 309–312
Vapors and Gases, Density of 70
Velocity of Flow Through Pipes 163
Velocity of Water in Ditches 160–161
Venturi Meter 114
Vertical Air Compressors, Single and Duplex 45–46
Vertical Boiler Feed or Pressure Pumps 349
Vertical Duplex Boiler Feed or Pressure Pumps 351
Vertical Duplex Light Service Pumps 364
Vertical Light Service Pumps 362
Vertical Piston Pumps 276
Vertical Sinking Pumps 365
Vertical Vacuum Pumps 372
Viscolizer 397
Viscosity of Oils 452–479
V. Notch Weir 114
Voltage 230
Volumes of Dry Saturated Steam at High Vacuums 176

Volumes of Free Air 84
Volumes, Mean Pressure and Temperature of Compressed Air 69
Volumetric Efficiency of Compressors 14–19
Volumetric Efficiency of Pumps 294
Volute Pumps 94

W

Water, Flow in Flumes 164
Water, Flow Through Pipes 163
Water, Hot, Handling of 287
Water for Irrigation 158
Water Horse Power 296
Water, Measurement of 110
Water Piston and Plunger 312
Water Rates of Turbo-Generators 180
Water, Relative Quantities 152
Water Seal, Open Pot 313
Water, Velocity in Ditches 160–161
Water Works Pumps 316
Weight of Air 85
Weight of Brass Tubing 218–225
Weights of Materials 439–440
Weights and Specific Gravity of Liquids 440
Weights and Specific Gravity of Metals 438
Weir Box 111
Weir Table 112–113
Well Pumps 342, 377
Wet Vacuum Power Pumps 389
Wet Vacuum Pumps 319
Wet Vacuum Pumps, Crank and Fly Wheel 398, 401
Wire Gauge Standards 418
Work of Adiabatic Compression 5
Work of Isothermal Compression 7
Work of Two Stage Compression 10
Wrought Iron Pipe 414
Wrought Iron Pipe, Extra Strong 415
Wrought Iron Pipe, Double Extra Strong 415



PLANT OF THE UNION STEAM PUMP COMPANY, BATTLE CREEK, MICHIGAN, U. S. A.

Manufacturers of Pumping Machinery for

Every Possible Service, from Municipal Water-Works to Country-Home Water-Supply.

FOREWORD

Authentic information on the proper selection, installation and operation of Pumping Machinery, Air Compressors and Condensers, has long been the source of constant search by engineers, architects and those interested in either the theory or practical application of this class of machinery.

Thirty successful years in the manufacture of this product has brought to us the realization of the need of a practical and condensed collection of this data, and we have, for the convenience of our friends, compiled this Union Engineering Handbook devoted to the theory and practice in design and use of Air Compressors, Centrifugal Pumps, Condensers, Steam and Power Pumps.

This book is presented as typical of the engineering service extended by—

UNION STEAM PUMP COMPANY,

Battle Creek, Michigan.

Air Compressors

SECTION ONE

Compressed Air and Air Compressors

Scarcely an industry exists that does not utilize compressed air in some manner. Second only to electricity in the extent and diversity of application, compressed air is one of the most important factors in every phase in the art of manufacture. The rapid development of compressed air appliances has brought about economical results that are reflected in every field of industry. As the economical application of compressed air is wholly dependent upon its economical production, it is apparent that the modern air compressor must embody every refinement in design and construction.

The cost of producing compressed air involves three separate items: first, interest and depreciation on the amount invested in compressed air equipment; second, operating cost; third, maintenance or upkeep cost. To minimize the cost of production, it is necessary to minimize each and every one of the above items.

As all three items depend upon the design and construction of the air compressor, good judgment and experience recommend as the best investment the purchase of a strictly high grade compressor, commanding a fair price which is a true measure of its value and which covers a construction insuring the lowest operating and upkeep cost. In Union Air Compressors will be found these necessary requirements.

The selection of the type of compressor depends entirely upon local conditions. Where steam is available, a steam driven unit is most desired. The steam cylinder constitutes a very efficient steam engine, and the power is transmitted direct, eliminating transmission losses and saving the expense of belts, shafts, pulleys, etc. On the other hand, there are numerous cases where a belt driven machine is far cheaper to operate, and the purchaser is always best competent to judge which type is the more desirable.

Principles of Compressed Air

In order to obtain an idea of the subject of air compression, there are certain underlying principles and laws that should be reviewed. On the following pages are given the basic laws and formulae for air compression that must be recognized when studying this subject.

Boyle's Law: At constant temperature, the volume of a gas is proportional to the absolute pressure or $PV = P_1 V_1$ in which

P = Initial absolute pressure in pounds per square inch.

V = Initial volume in cubic feet.

P_1 = Final absolute pressure in pounds per square inch.

V_1 = Final volume in cubic feet.

This law expresses the fact that if the pressure on a certain volume of gas is doubled, the volume will be one-half the original volume (if the temperature is constant), or conversely, if at constant temperature, the pressure is reduced one-half, the volume will be doubled.

Charles' Law: At constant volume, the pressure of a perfect gas is directly proportional to the absolute temperature or at constant pressure the volume is directly proportional to the absolute temperature or:

$$\frac{P}{T} = \frac{P_1}{T_1} \text{ and } \frac{V}{T} = \frac{V_1}{T_1}$$

in which T and T_1 are initial and final absolute temperatures in degrees Fah.

Combining Charles' and Boyle's laws, we have the formula

$$\frac{P\,V}{T} = \frac{P_1\,V_1}{T_1}$$

Joules' Law: When a perfect gas expands doing no external work, the temperature remains constant. For example in the equation

$$\frac{P\,V}{T} = \frac{P_1\,V_1}{T_1}$$

If $T = T_1$ we have $P\,V = P_1\,V_1$, which is the law of expansion of a perfect gas.

Specific Heat—The Specific heat of a substance is the amount of heat (B. T. U.) that is required to raise the temperature of a pound of the substance through 1° Fah.

Specific Heat at Constant Volume C_v

In the equation $\frac{PV}{T} = \frac{P_1 V_1}{T_1}$

If $V = V_1$ then we have $\frac{P}{T} = \frac{P_1}{T_1}$

which is the law of Charles. Suppose we have a certain volume

of air contained in a closed receptacle, and the temperature is raised 1° Fah. The pressure is thereby raised according to the above law, and the intrinsic energy of the air increased. No work is done, however, because work equals pressure multiplied by distance, and by our supposition the latter factor is zero. At constant volume, then, the specific heat of air is the amount of heat (B. T. U.) that is required to raise the temperature of one pound of the air through 1° Fah., the volume being kept constant as above. C_v for air is found by experiment to be .1685.

Specific Heat at Constant Pressure C_p

Assume in this instance that we have a vertical cylinder containing a quantity of air, and resting on the air is a frictionless piston of constant weight, or pressure P. If the air is heated, the volume will increase, moving the piston outward, and external work is performed. At constant pressure, then, the specific heat of air is the amount of heat (B. T. U.) that is required to raise the temperature of one pound of the air through 1° Fah., if the air is allowed to expand against a constant pressure. Therefore $C_p = C +$ the heat equivalent of external work, and for air has been found to be .2375. C_p and C_v are measured in B. T. U's., so as to obtain their equivalent in foot pounds, it is necessary to multiply by 778, and the products for convenience of calculation are called K_p and K_v.

Getting back to our assumption of the cylinder and piston and assuming further that we have (W) pounds of air; in order that external work be done, and the temperature raised 1° Fah., it is necessary that W $(C_p - C_v)$ thermal units of heat be applied, or W $(K_p - K_v)$ foot pounds of work. In order to raise the temperature T degrees, W $(K_p - K_v)$ T foot pounds of work must be done on the air. Since work is equal to pressure through volume, we have

$$\text{Work} = P\,V = W \times (K_p - K_v) \times T°$$

or assuming $(K_p - K_v) = R$, we have the formula:

$$P\,V = W\,R\,T. \qquad (1)$$

Theoretically air may be compressed in two ways:— Adiabatically or Isothermally.

Adiabatic Compression

Adiabatic compression of air is compression without loss of heat. Consider for example a perfectly insulated air

cylinder and piston having full charge of air between the piston and cylinder head. As the piston moves, the volume of air becomes smaller, and the temperature rises, the former in inverse proportion to the absolute pressure exerted, and the latter equivalent to the amount of work done. Under these conditions the air at the end of the compression will retain all of the heat so produced, and this particular compression is called adiabatic. In actual practice such conditions of compression are impossible.

In adiabatic compression the law $\frac{P}{P_1} = \frac{V_1}{V}$ is not followed strictly, because as the temperature rises unchecked, it reacts on the air being compressed to increase the volume. Therefore to write an expression for adiabatic compression, it is necessary that $\frac{V_1}{V}$ be increased by an amount equivalent to the amount of external work done on the air by heat reaction during compression. It has been shown by various authorities on heat or thermodynamics that

$$\frac{P}{P_1} = \frac{(V_1)^n}{(V)^n} \text{ in which } n = \frac{C_p}{C_v} = \frac{.2375}{.1685} = 1.41$$

for air holds nearly true.

Work of Adiabatic Compression

Figure 1 shows the theoretical indicator card of an air cylinder having no clearance. The total work done is equal to the work of compression shown by the area under the curve B C, plus the work of expulsion of the air from the cylinder shown by the area $P_2 V_2$, minus the work done on the piston by the inlet air shown by the area $P_1 V_1$. Then calling Q the total amount of work

$$Q = 498.7\ P_1 V_1 \left(\left[\frac{P_2}{P_1} \right]^{.29} - 1 \right) \qquad (2)$$

and the horse power required to compress 1 cubic foot of free air per minute adiabatically is

$$\text{H. P.} = \frac{1}{4.5}\left(\left[\frac{P_2}{P_1}\right]^{.29} - 1\right) \qquad (3)$$

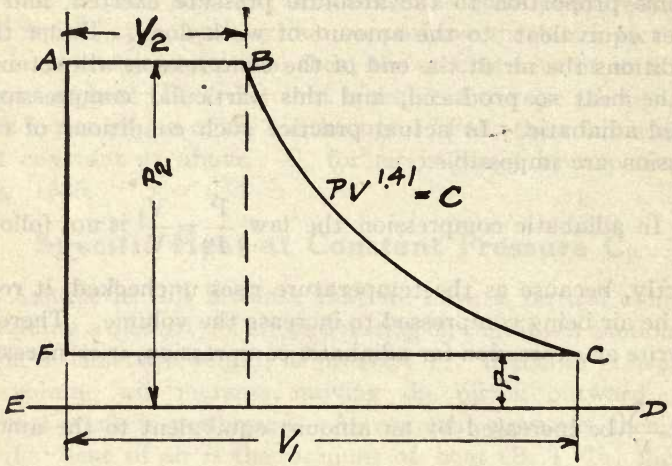


Fig 1

EXAMPLE: What horse power will be required to compress adiabatically 1 cubic foot of free air at sea level to 100 pounds gauge pressure?

SOLUTION: Inserting the above values in equation 3:

$$\text{H. P.} = \frac{1}{4.5}\left(\left[\frac{114.7}{14.7}\right]^{.29} - 1\right)$$

$$= \frac{1}{4.5}(1.81 - 1)$$

$$= .18$$

Isothermal Compression

Isothermal compression is compression at constant temperature. In other words this is compression wherein all heat is removed by some form of cooling device as fast as it is produced. The relation then existing between pressure and volume at any instant is shown by the equation:

$$P_1 V_1 = P_2 V_2 = C \qquad (4)$$

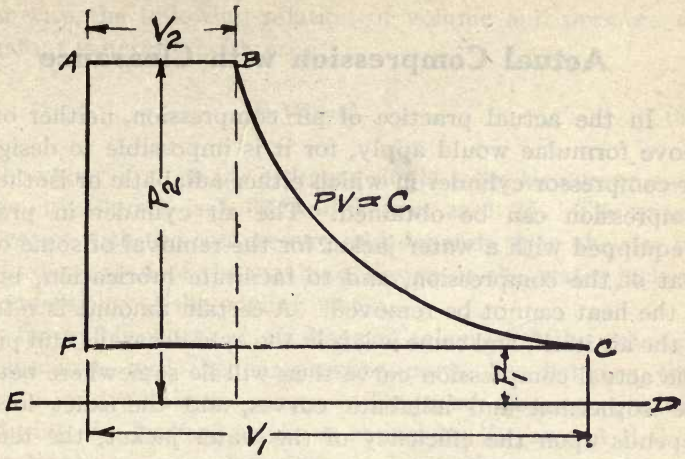


Fig. 2.

Work of Isothermal Compression

Fig. 2 is the theoretical indicator card of isothermal compression in a cylinder having no clearance. The compression begins as before at absolute pressure P_1 and volume V_1, and ends at P_2 and V_2. The total work Q in foot pounds done on the air is equal to the algebraic sum of the work of the compression, expulsion and the work done by the intake air, and is shown in the following equation:

$$Q = 144\, P_1\, V_1 \log_e \left[\frac{P_2}{P_1}\right] \qquad (5)$$

and the horse power required to compress 1 cubic foot of free air per minute isothermally is

$$\text{H. P.} = \frac{1}{15.6} \log_e \left[\frac{P_2}{P_1}\right] \qquad (6)$$

EXAMPLE: What horse power will be required to compress isothermally 1 cubic foot of free air at sea level to 100 pounds gauge pressure?

SOLUTION: Inserting the above values in equation 6,

$$\text{H. P.} = \frac{1}{15.6} \log_e \left[\frac{114.7}{14.7}\right]$$

$$= \frac{1}{15.6} \log_e 7.8$$

$$= \frac{1}{15.6} \times 2.05$$

$$= .132$$

Actual Compression with Clearance

In the actual practice of air compression, neither of the above formulae would apply, for it is impossible to design an air compressor cylinder in which either adiabatic or isothermal compression can be obtained. The air cylinder in practice is equipped with a water jacket for the removal of some of the heat of the compression, and to facilitate lubrication, but all of the heat cannot be removed. A certain amount is retained in the air itself, and some is left in the cylinder walls and piston. The actual compression curve then will lie somewhere between the isothermal and adiabatic curves, and the exact location depends upon the efficiency of the water jacket, the temperature of the circulating water, etc.

Also in the actual cylinder there is a certain amount of lost space or clearance in the ends of the cylinder between the piston and the heads, and around the valves, all of which has its effect on the shape of the indicator card. This clearance at the end of the piston is filled with air at the discharge pressure and temperature. As the piston recedes, the clearance air expands, doing work on the piston, and finally the re-expanded air occupies part of the volume of the cylinder behind the piston. No air can be drawn into the air cylinder until the pressure inside falls below that of the atmosphere.

In the compression of a perfect gas receiving heat in some

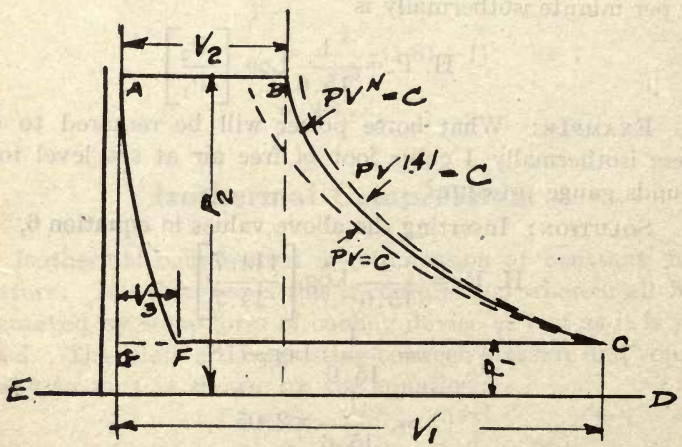


Fig. 3

regular way, the following relation of volume and pressure at any instant generally holds true:

$$P V^n = C \qquad (7)$$

The value of n on the ordinary single stage air compressor as given by Church is 1.33, and Unwin as 1.25. The exact value varies with the compressor, and depends upon the size of the cylinder, speed of the machine, design of the water jacket, and temperature of the cooling water.

In figure 3 is shown an air diagram taken from the cylinder with clearance, and the compression curve lies between the isothermal and adiabatic curves. The total amount of work done during the forward stroke is shown by the area A B C G (q), but by the re-expansion of clearance air there is an amount of work Q^1 returned to the receding piston, shown by the area A F G. Therefore the net amount of work done by the piston is shown by the area A B C F or Q, and its value is given in the expression

$$Q = \frac{P_1(V_1 - V_3)\, n}{n-1}\left(\left[\frac{P_2}{P_1}\right]^{\frac{n-1}{n}} - 1\right) \qquad (8)$$

$(V_1 - V_3)$ is the net amount of air drawn into the cylinder. The horse power required to compress one cubic foot of free air per minute is shown by the following equation:

$$\text{H. P.} = \frac{n}{15.6\ (n-1)}\left(\left[\frac{P_2}{14.7}\right]^{\frac{n-1}{n}} - 1\right) \qquad (9)$$

EXAMPLE: What horse power will be required to actually compress one cubic foot of free air at sea level to 100 pounds gauge pressure?

SOLUTION: Inserting these values in equation 9.

$$\text{H. P} = \frac{1.25}{15.6 \times .25}\left(\left[\frac{114.7}{14.7}\right]^{\frac{1.25-1}{1.25}} - 1\right)$$

$$= .320 \quad (7.8^{.2} - 1)$$

$$= .320 \quad (1.51 - 1)$$

$$\text{H. P.} = .163$$

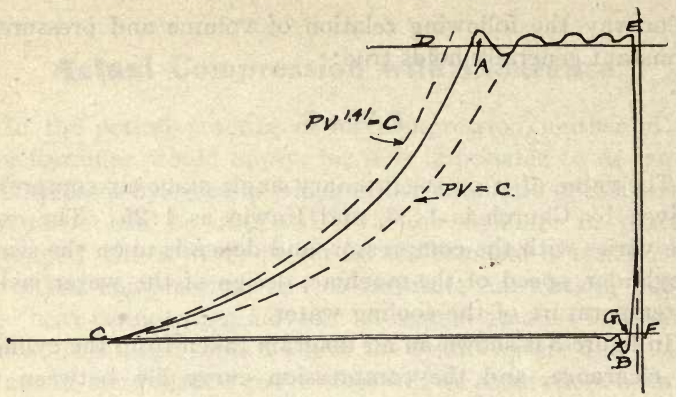


Fig. 4.

Figure 4 shows the actual air indicator card taken from a single stage air compressor having disc valves on the inlet and discharge. The areas A and B represent the amount of work necessary to open the discharge and the inlet valves, and G F is the volume occupied by the re-expanded clearance air. The volume lying between the suction line and the atmospheric line is the energy expended to fill the cylinder with air.

Two Stage Air Compression

It is evident now that isothermal compression requires the expenditure of the least amount of power. As before shown this form of compression is impossible in practice, but an approach to it is realized by compression in stages, and cooling the air between each stage. In this way isothermal compression is partially realized as will be seen later.

In two stage compression the air is drawn from the atmosphere into the first or low pressure cylinder, and there compressed up to a certain pressure. It is then discharged through an intercooler where the temperature is reduced by circulating water, and then drawn into the second or high pressure cylinder where the compression is continued up to the desired terminal pressure.

Work of Two Stage Air Compression

In a two stage compressor it is customary to proportion the cylinders so that the work is equally divided between the two.

In the following it is assumed that the work is the same in each cylinder, and further that the temperature of the air

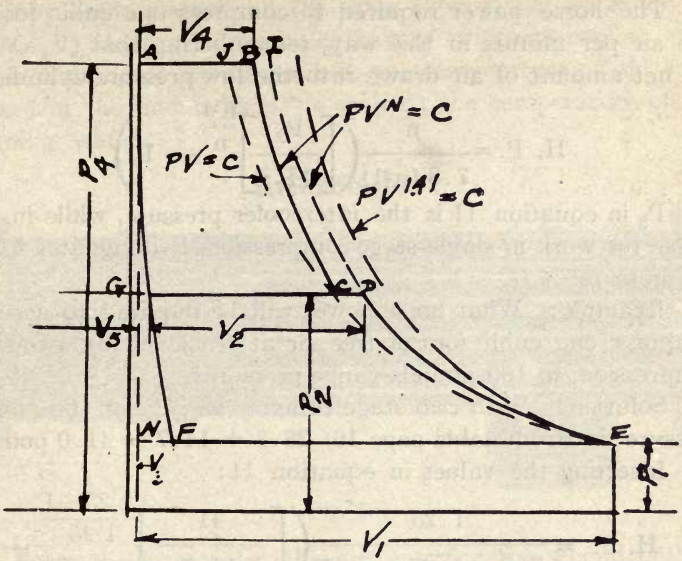


Fig. 5

after passing through the intercooler is the same as the atmosphere.

Figure 5 shows the cycle of operation in a two stage compressor. A volume V_1 of air under pressure P_1 is drawn into the low pressure cylinder, and there compressed to volume V_2, and pressure P_2. The air is cooled and the volume is reduced to that shown by GC, which is equivalent to the volume obtained in isothermal compression from P_1 to P_2. The high pressure cylinder then receives the air and compresses it up to the pressure P_4 and volume V_4. The curve of compression follows the broken line EDCB.

If the air was compressed in a single stage compressor from P_1 to P_4, the curve of compression would be E I ($P V_n = C$) and the work done shown by area AIEF. The work done by two stage compression is shown by the area ABCDEF, and the saving realized by staging is shown by the area BIDC.

If Q_1 and Q_2 equal work done in foot pounds, to compress air in the low pressure and high pressure cylinders respectively, and Q = total work of compression, the value of Q then is

$$Q = 288 \frac{P_1 (V_1 - V_3) n}{n-1} \left(\left[\frac{P_2}{P_1} \right]^{\frac{n-1}{n}} - 1 \right) \qquad (10)$$

The horse power required to compress one cubic foot of free air per minute in this way, remembering that $(V_1 - V_3)$ is the net amount of air drawn into the low pressure cylinder, is

$$\text{H. P.} = \frac{n}{7.8\,(n-1)}\left(\left[\frac{P_2}{14.7}\right]^{\frac{n-1}{n}} - 1\right) \qquad (11)$$

P_2 in equation 11 is the intercooler pressure, while in formulae for work in single-stage compression, P_2 designates terminal pressure.

Example: What horse power will be required to actually compress one cubic foot of free air at sea level, by two stage compression, to 100 pounds gauge pressure?

Solution: With two stage compression, P_2 for 100 pounds pressure is (from table page 16) $26.3 + 14.7 = 41.0$ pounds.

Inserting the values in equation 11:

$$\text{H. P.} = \frac{1.25}{7.8\ \ (1.25\ \ - 1)}\left(\left[\frac{41}{14.7}\right]^{\frac{1.25-1}{1.25}} - 1\right)$$

$$= \quad .64\ \ (2.78^{.2} - 1)$$

$$= \quad .64 \times .227$$

$$= \quad .145$$

Multi-Stage Compression and Its Advantages

Theoretically there is a gain in multi-stage compression, whatever the pressure. However, with low pressures, the saving is so small as to be offset by the additional expense involved, and the unavoidable mechanical losses in the operation of the additional mechanism. Experience has fixed from 80 pounds to 100 pounds gauge as maximum terminal pressures, which can be best attained with a single stage compressor; and for pressures from 80 pounds up, multi-stage compression in two, three and four-stage compressors is employed.

In multi-stage air compressors correctly designed, the cylinder ratios are such that the final temperatures, and mean effective pressures are equal in all cylinders, and all pistons are, therefore, equally loaded. The air compressed in the first cylinder with a pressure determined by the cylinder ratio, is discharged through the discharge valves to an intercooler where it is split up into thin streams passing over cold surfaces. Modern practice involves a nest of tubes through which cold water circulates, and over and between which the stream of air passes, complete breaking-up and subdivision of the stream being se-

cured by baffle plates, and the tubes themselves (see figure 6). A properly designed intercooler having sufficient cooling area for the volume of air may reduce the temperature of air compressed in the first cylinder to at least the temperature of the outgoing water.

Intercooler

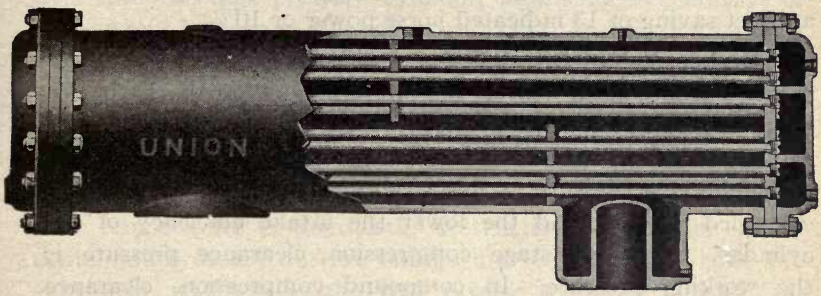


Fig 6.

From the intercooler, this air, entering the second or high pressure cylinder, is compressed to a higher pressure, and again reaches a temperature about the same as that attained in the first cylinder. In two stage machines, this air will be discharged directly to the receiver without further cooling unless conditions are such as to render necessary the use of an aftercooler.

The principal advantages of multi-stage compression are: Reduced power, higher volumetric efficiency, drier air, and better lubrication. These will be considered in order.

Reduced Power: The table on page 68 gives the percentage of work lost in the heat of compression in one, two, or three stages at various pressures. In these figures no account is taken of jacket cooling, nor is any allowance made for certain inevitable mechanical losses.

Taking a specific example, the saving by multi-staging is evident. Assuming a volume of compressed air equivalent to 100 effective horse power is to be delivered at a pressure of 100 pounds.

Referring to the table in column 2, the theoretical percentage of lost work in one stage compression is given at 36.7; but because there is bound to be some radiation of heat, the value of 36.7% will not be found in practice, and 30% may be assumed as a practical figure under average conditions. On this basis, it is found that to deliver 100 horse power in compressed air at 100 pounds pressure, by one stage compression, there will be required 130 indicated horse power. Looking

now at column 4 of the table, the percentage of loss in two stage compression at this pressure is found to be 16.9 per cent, which is very close to the figure found in practice. Applying this value, it is seen that to deliver the equivalent of 100 effective horse power in air at 100 pounds pressure, by two stage compression, about 117 indicated horse power will be required. In this case as between single and two stage compression, we have a direct saving of 13 indicated horse power or 10%.

Higher Volumetric Efficiency: Before free air can enter through the suction valves, the air remaining in the clearance space between piston and head, at the end of the stroke, must be expanded on the return stroke to atmospheric pressure. Evidently the higher the pressure in this clearance space, the greater this expanded volume, and the lower the intake efficiency of the cylinder. In single stage compression, clearance pressure is the working pressure. In compound compression, clearance pressure in each cylinder is terminal pressure in that cylinder, but this terminal pressure in the intake cylinder is low, generally not over 25 pounds, when the final working pressure is 100 pounds. The volumetric efficiency of a multi-stage compressor is higher for this reason, the clearance in the low pressure cylinder only, being in question.

Another reason for higher volumetric efficiency resulting from multi-stage compression is the fact that terminal pressures, and consequently terminal temperatures are lower than in single stage cylinders. The cylinder walls and more particularly the heads with the valves and ports which may be in them are therefore kept much cooler, and the entering air is not much heated by contact with these parts. A third element entering into the question of capacity is the reduced leakage in stage-compression cylinders through valves and past pistons and rods with the resultant loss of power. It is evident that the higher the pressure, the greater liability to leaking; and the smaller range of partly balanced pressures in multi-stage cylinders reduce this loss.

Drier Air: One of the greatest difficulties encountered in air power transmission has been the freezing of the moisture in the air, either in the pipe line, or at the exhaust ports of the air motors. One of the great advantages of multi-stage compression lies in the opportunity it affords for cooling the compressed air between stages to a temperature at which its moisture will be precipitated. Practically all of this condensation occurs in the intercooler; and herein appears the necessity for a design,

which will allow the air to pass over the tubes at a low velocity with a full opportunity for cooling on the tubes. The moisture in suspension is withdrawn through the drain pipe, which is provided at the lowest point of the intercooler. Unless the moisture is not withdrawn from the intercooler, the value of the latter as an air drier is lost; for the moisture is carried over into the high pressure cylinder, producing a condition of cutting and leakage in the valves and rings, and finally working into the pipe line. Aftercoolers are in some instances as important as intercoolers in removing the moisture.

Better Lubrication: If air be compressed in a single cylinder, from atmosphere, and a temperature of 60° Fah. to a final pressure of 100 pounds, the maximum temperature will be 484° Fah. This temperature is manifestly destructive to common lubricants, and ordinary oils are burned into a solid gritty coke-like substance, which gives the very reverse to proper lubrication, unless proper cooling devices are employed to keep the parts cold. This carbon deposit collecting in ports and valves may so obstruct and clog them as to cause leakage and throw an additional load on the compressor. If, however, the same volume of air be compressed in the low pressure cylinder to a pressure of 25 pounds, the highest temperature which can be reached is only 233°., a heat which will not leave a deposit or destroy the lubricating qualities of good oils such as should be used in air compressor work. This air passing through the intercooler will be brought back to approximately the original temperature of 60°, and compressed in the second stage, or high pressure cylinder, from 25 pounds to 100 pounds. Here the maximum temperature will be little, if any in excess of that in the first stage cylinder, since the heat of compression is a function of the number of compressions, and is almost wholly independent of initial pressure. In multi-stage compressors, therefore, the conditions of temperatures are seen to be most conducive to thorough lubrication of the pistons and valves tending towards durability and tightness of the working parts with sustained efficiency of the machine.

Pressures for Stage Compression

Single stage compression is used for pressures up to 100 pounds. Two stage compression for pressures from 80 pounds to 500 pounds, three stage compression for pressures 500 pounds to 1500 pounds, and four stage compression for pressures above 1500 pounds.

Correct Cylinder Ratios for Two Stage Compression.

The correct ratio of cylinders is obtained by the following formula:

or 2 stage compression,

$$r = \sqrt{\frac{P_3}{P}} \qquad (12)$$

In which r = ratio of cylinders.

P_3 = Absolute terminal pressure in pounds per square inch.

P = Atmospheric pressure in pounds per square inch.

Thus in two stage compression, we extract the square root of the number of atmospheres to be compressed. This proportion of cylinder volumes divides the work equally between the different stages.

The intercooler pressure (P_1) in a two stage compressor is obtained by the following formula:

$$P_1 = P \times \sqrt{\frac{P_3}{P}} - P \qquad (13)$$

In which P_1 = Intercooler pressure between first and second stages.

The following table gives the correct cylinder ratio and intercooler pressure in two stage compression for gauge pressures from 50 to 500 pounds per square inch.

Gauge Pressure Pounds	Absolute Pressure, Pounds	Number of Atmospheres	Correct Ratio of Cylinder Volumes	Intercooler Gauge Pressure	Gauge Pressure Pounds	Absolute Pressure, Pounds	Number of Atmospheres	Correct Ratio of Cylinder Volumes	Intercooler Gauge Pressure
50	64.7	4.40	2.10	16.2	200	214.7	14.60	3.82	41.4
60	74.7	5.08	2.25	18.4	210	224.7	15.28	3.91	42.8
70	84.7	5.76	2.40	20.6	220	234.7	15.96	3.99	44.0
80	94.7	6.44	2.54	22.7	230	244.7	16.64	4.08	45.3
90	104.7	7.12	2.67	24.5	240	254.7	17.32	4.17	46.6
100	114.7	7.80	2.79	26.3	250	264.7	18.00	4.24	47.6
110	124.7	8.48	2.91	28.1	260	274.7	18.68	4.32	48.8
120	134.7	9.16	3.03	29.8	270	284.7	19.36	4.40	50.0
130	144.7	9.84	3.14	31.5	280	294.7	20.04	4.48	51.1
140	154.7	10.52	3.24	32.9	290	304.7	20.72	4.55	52.2
150	164.7	11.20	3.35	34.5	300	314.7	21.40	4.63	53 4
160	174.7	11.88	3.45	36.1	350	364.7	24.80	4.98	58.5
170	184.7	12.56	3.54	37.3	400	414.7	28.20	5.31	63.3
180	194 7	13 24	3.64	38.8	450	464.7	31.60	5.61	67.8
190	204.7	13.92	3.73	40.1	500	514.7	35.01	5.91	72.1

Air Compression at Altitudes

If a compressor is operated at a greater altitude than sea level (14.7 pounds per square inch), the intake air pressure will be proportionately less, and additional work is imposed upon the compressor.

The capacity of a given compressor is less at higher altitudes than at sea level, because of the diminished density of the intake air.

Volumetric efficiency is also less at altitudes, due to the fact that the clearance air expands to the atmospheric pressure and consequently when expanding occupies a larger volume of the cylinder.

The table on page 70 gives the multipliers for compression at altitudes.

Mechanical Efficiency

The mechanical efficiency of a steam driven compressor is equal to the air indicated horse power divided by the steam indicated horse power or

$$E_m = \frac{\text{Air Indicated H. P.}}{\text{Steam I. H. P.}} \qquad (14)$$

and the mechanical efficiency of a power driven air compressor is

$$E_m = \frac{\text{Air Indicated H. P.}}{\text{Brake H. P. Delivered to compressor shaft.}} \qquad (15)$$

This efficiency depends on the mechanical construction of the compressor and the lubrication. It varies from 80% to 92%.

Compression Efficiency

Compression efficiency is the ratio of the theoretical Horse Power required to compress an amount of air to that actually required or

$$E_c = \frac{\text{Theoretical Horse Power.}}{\text{Actual Horse Power.}} \qquad (16)$$

This efficiency depends upon the water jacket and cooling devices, and it is principally to increase compression efficiency that compound compression is employed. To determine the compression efficiency, the isothermal curve is plotted on the air card figure 7, starting of course at the beginning of the stroke and ending at the theoretical delivery line, or thermal pressure line. The area AFDE, this divided by the area ABDC of the actual card, is the compression efficiency. Actual compression curves will follow the adiabatic curve quite closely as the water jacket has little effect other than to facilitate lubrication.

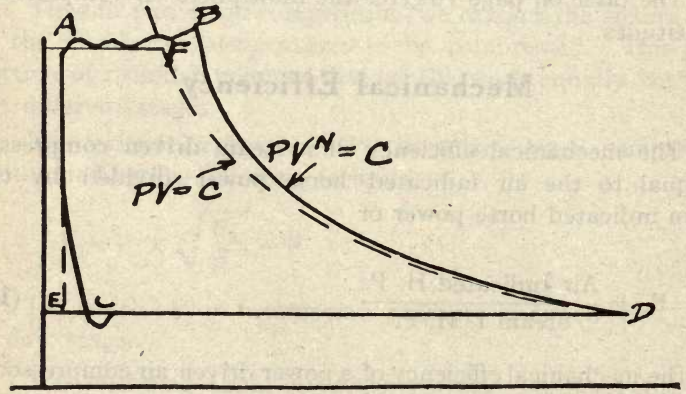


Fig. 7.

Displacement

Air compressors are always rated according to displacement, that is the volume displaced by the net area of the compressor piston.

Capacity

The capacity should be expressed in cubic feet per minute of free air at intake temperature, and at the pressure of dry air at the suction.

Volumetric Efficiency

Volumetric efficiency is the ratio of the actual number of cubic feet of free air compressed per unit of time to the number of cubic feet of piston displacement during that time or,

$$E_v = \frac{\text{Actual cubic feet of free air per minute.}}{\text{Cubic feet of piston displacement per minute.}} \quad (17)$$

On the indicator diagram, the observed volumetric efficiency is $\frac{CD}{ED}$ (fig. 7). Volumetric efficiency depends upon the clearance volume in the air cylinder. If there were no clearance spaces in the cylinder, the volumetric efficiency would be 100%. The greater the clearance volume, the greater the volume of the cylinder occupied by the expanded clearance air. Volumetric efficiency depends upon the terminal pressure. The higher the terminal pressure of the air in the cylinder, the greater will be the volume occupied by the expanded air of the clearance spaces. This means that as the terminal pressure is increased, the volumetric efficiency decreases.

Volumetric efficiency depends upon the temperature and pressure of the intake air. Since volumetric efficiency refers to free air at 14.7 pounds and 60° Fah., then every change of temperature and pressure of intake has its effect upon the volumetric efficiency of the compressor. For example, let us suppose the temperature of the intake air of the compressor is 60° Fah., and the atmospheric pressure 14.7 pounds, or in other words it is actual free air that is drawn into the cylinder. If the compressor has a displacement of 100 cubic feet of free air per minute, and actually delivers 85 cubic feet of free air per minute at 100 pounds pressure, the volumetric efficiency is

$$E_v = \frac{85}{100} = 85\%$$

Now if the temperature of the intake air is raised to 65° Fah, the other conditions of operation remaining the same, the compressor will still deliver 85 cubic feet of free air per minute, but owing to its higher temperature, a smaller weight or mass of air will be taken into the cylinder according to the law of Charles. 85 cubic feet of air at 60° Fah. and 14.7 pounds pressure is equivalent to 84 cubic feet at 65° Fah., and 14.7 pounds pressure. Under these conditions the volumetric efficiency decreases

$$E_v = \frac{84}{100} = 84\%.$$

This shows that there is a loss of 1% in volumetric efficiency for every 5° rise in temperature of the intake air.

Volumetric efficiency is also affected by changing the atmosphere or intake pressure, the temperature remaining constant. To show this, assume that the compressor, referred to above, were removed to an altitude of 5000 feet, where the intake air is at 12.20 pounds pressure, and 60° Fah. Now 85 cubic feet of this air is equivalent to 70 cubic feet of free air at sea level and the volumetric efficiency is

$$E_v = \frac{70}{100} = 70\%.$$

In the selection of air compressors, it should be borne in mind that they are always rated according to the piston displacement, and due allowance must be made for the volumetric efficiency. The volumetric efficiency of the average compressor varies from 70% to 95% according to the size and conditions of operation.

Compressor Installation and Operation.

The large majority of instances of unsatisfactory operation of air compressors eminates from the improper installation in the first place, and continued negligent operation and disregard of the compressor manufacturer's instructions in the second place. An air compressor is looked upon by many engineers as a machine that can be tucked away in an out-of-way place, and left to itself without any attention.

An air compressor, like any piece of machinery, requires some attention for successful operation, and if the operator, before erecting a compressor, will spend a little time familiarizing himself with the practical principles of an air compressor, the biggest majority of the cases of trouble will be eliminated. A careful study of the compressor manufacturer's instructions will enlighten the operator on this subject.

Location.

An air compressor should be installed in a place which is clean and cool, and ample space be provided all around the compressor for cleaning and inspecting. Too often compres-

sors are pushed off in a corner where it is an impossibility to get around them. Locations of compressors in boiler rooms, near coal piles, or other places where there is liable to be an accumulation or a settling of dust and dirt, should be avoided.

Foundations.

Air compressor foundations depend of course upon the size and type of air compressor, as well as the nature of the soil.

An air compressor, like an engine, requires a rigid foundation to prevent any vibration. The value of a foundation made of good materials, and well built, will be readily understood. The slight difference in cost between the best and inferior materials and workmanship will save future annoyance and expense. The materials used in the foundation depend somewhat upon local conditions, but it is advisable to use cement concrete, as this material furnishes an excellent foundation at a comparatively small cost.

The following mixture is recommended for concrete foundations: One part Portland cement, three parts coarse sand and six parts broken stone.

Allow the foundation to stand at least a week after it is completed, before placing the compressor upon it.

With each compressor, the manufacturer sends out a detailed foundation plan, and the foundation can be laid out and built from this plan before the compressor is received.

Air Inlet Piping.

It has already been shown that an increase of 5° Fah. in the temperature of the intake air is accompanied by a decrease of 1% for volumetric efficiency, which means that as the intake air temperature increases, the free air capacity of the compressor decreases, and the same amount of power is expended as though the full capacity of the compressor was being realized.

To get the best results, the air intake should be piped to the outside of the building, some 8 or 10 feet above the ground level, or above the roof. Dust and dirt must by all means be prevented from entering the compressor, as it will cut and wear the cylinder surfaces, as well as the valve seats, and cause all manner of trouble. The intake opening should be hooded to keep out rain, and carefully screened to eliminate dust and dirt.

The intake pipe should be at least the size of the inlet on the compressor, and wherever the line is of any length, it is advisable to increase this size to avoid frictional losses.

Discharge Piping

The discharge pipe should be at least the diameter of the discharge opening in the air cylinder, and contain as few bends as possible. The discharge pipe should be carried the full size into the receiver.

Air Receivers

The functions of an air receiver are (1) to create a cushion and eliminate the compressor pulsations in the pipe line; (2) to serve as a storage of power; (3) to cool the air and precipitate any oil or moisture in entrainment; (4) to eliminate friction losses that would occur, if cooling were effected in the pipe lines. The receiver should therefore be located in a cool place, and as close as possible to the compressor. The receiver fittings should include pressure gauge, safety valve and blow-off cock.

Lubrication

Bearings in modern air compressors, the main bearings as well as the connecting rod and crosshead bearings, are usually lubricated by means of the splash system.

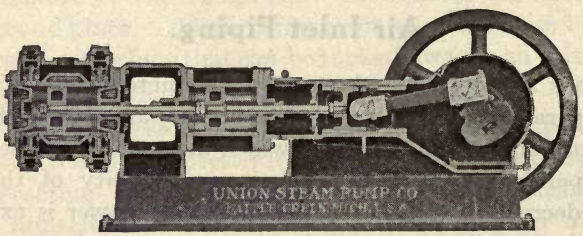


Fig 8.

The sectional view shown herewith, illustrates an enclosed type air compressor lubricated by the splash system. The frame forms with its covers a closed chamber, with a quantity of oil in the basin below the crank, into which the crank and connecting rod dip at each revolution. The motion of these parts splashes the oil to every bearing, and insures copious lubrication. By means of a well designed settling chamber,

any sediment or abrasive material, which may accumulate in the oil, is returned to the bottom of the frame, and prevented from being carried through the bearings. Close fitting cast iron covers prevent any leakage from the frame.

Air Cylinder: The air cylinder lubrication is by far the most vital point in air compressor lubrication, and as a rule it seems to be the least understood. In order to appreciate the necessity of proper cylinder lubrication, it is advisable to consider the conditions that have to be met.

The compression of air results in a rise in temperature according to the equation on page 3. For adiabatic compression of air, the temperature and pressure relations are expressed by the formula:

$$\frac{T_1}{T} = \left[\frac{P_1}{P}\right]^{\frac{n-1}{n}} = \left[\frac{P_1}{P}\right]^{.29} \qquad (18)$$

and $$T_1 = T\left[\frac{P_1}{P}\right]^{.29}$$

Where T and T_1 are the initial and final absolute air temperatures, and P and P_1 the initial and final absolute pressures, therefore the temperature of the air at discharge from the cylinder is dependent not only upon the pressure, but on the temperature of the intake air. For example, assume a single stage compressor operating at sea level at an atmospheric temperature of 60° Fah., and discharging against 70 pounds pressure. The final temperature then is:

$$T_1 = 521° \left[\frac{84.7}{14.7}\right]^{.29} = 866° \text{ absolute}$$

or 405° Fah.

TABLE 1. CYLINDER TEMPERATURES AT END OF PISTON STROKE.

Final Pressure of Air Lb. Gage	Final Temperature Deg. F.		Final Pressure of Air Lb. Gage	Final Temperature Deg. F.	
	Single-Stage	Two-Stage		Single-Stage	Two-Stage
10	145	...	100	485	243
20	207	...	110	507	250
30	255	...	120	529	257
40	302	...	130	550	265
50	339	188	140	570	272
60	375	203	150	589	279
70	405	214	200	672	309
80	432	224	250	749	331
90	459	234	...	...	...

This calculation does not take into consideration jacket cooling, or heat radiation, so in practice the actual discharge temperature will be slightly less. The foregoing table gives the cylinder temperature, in single and two stage compression for pressures from 10 pounds up to 250 pounds gauge.

For successful lubrication of the air cylinders, it is necessary to use oil which reduces the friction to a minimum, and eliminates carbonization as much as possible. Carbonization is generally caused by using a poor grade of oil, such as steam cylinder oil, which is easily decomposed by the heat of compression, or the use of too great a quantity of oil, or the failure to properly screen the intake pipe of the compressor, thus permitting dust and foreign matter to enter the air cylinder.

Air cylinder oil should be a medium body pure mineral oil of either a paraffin-base or asphaltic-base.

With the paraffin-base oil, any carbon deposit is very adhesive, and of a hard flinty material; while with the asphaltic-base oil, the carbon deposit is of a light fluffy nature, and easily cleaned out.

The following tables published by the Compressed Air Society will serve as a guide to specify the qualities to be possessed by an oil for air cylinder lubrication.

The average range of figures are recommended for single stage compression up to 100 pounds pressure, and for two stage air compression for higher pressures in which the air is cooled between stages so that the maximum terminal temperature is not in excess of that due to a pressure of 100 pounds for a single stage compressor.

TABLE II. PHYSICAL TESTS OF PARAFFIN-BASE OILS.

	Minimum	Average	Maximum
Gravity, Baumé	28 to 32 deg.	25 to 30 deg.	25 to 27 deg.
Flash point, open cup	375 to 400 deg. F.	400 to 425 deg. F	425 to 500 deg. F
Fire point	425 to 450 deg. F	450 to 475 deg. F	475 to 575 deg. F
Viscosity (Saybolt) at 100 deg. F.	120 to 180 sec.	230 to 315 sec.	to 1500 sec.
Color	Yellowish	Reddish	Dark red to green
Congealing point (pour test)	20 to 25 deg. F	30 deg. F.	30 to 45 deg. F

TABLE III. PHYSICAL TESTS OF ASPHALTIC-BASE OILS

	Minimum	Average	Maximum
Gravity, Baumé	20 to 22 deg. F	19.8 to 21 deg.F	19.5 to 20.5 deg. F.
Flash point, open cup	305 to 325 deg. F	315 to 335 deg. F	330 to 375 deg. F
Fire Point	360 to 380 deg. F	370 to 400 deg. F	385 to 440 deg. F
Viscosity (Saybolt) at 100 deg. F.	175 to 225 sec.	275 to 325 sec.	475 to 750 sec.
Color	Pale yellow	Pale yellow	Pale yellow
Congealing point (pour test)	0 deg. F.	0 deg. F.	0 deg. F.

Quantity of Lubricating Oil for Air Cylinders

The proper quantity of lubricating oil to be used in the air cylinders depends upon the viscosity of the oil, but it should be borne in mind that when the surface of the cylinder wall is once glazed over, little oil is required to lubricate the surfaces. The following table shows the approximate quantity of oil for air cylinder lubrication:

TABLE IV. QUANTITY OF AIR-CYLINDER LUBRICANT REQUIRED PER 10-HOUR DAY.

Size of Cylinder, Inches	Displacement per Minute Cu. Ft.	Piston Speed, Feet per Minute	Sq. Ft. of Cyl. Wall Swept by Piston	Drops of Oil per Minute	Drops of Oil per 10 Hours	Sq. Ft. Oiled per Drop	Pints of Oil per 10 Hours
6⅛x 6	57	330	440	1	600	440	.0375
8x 8	124	354	740	2	1200	370	.0750
10x10	228	416	1095	2	1200	548	.0750
12x12	371	470	1480	3	1800	493	.1125
14x15	561	525	1930	4	2400	483	.1500
16x15	730	525	2200	4	2400	550	.1500
18x15	930	525	2480	5	3000	496	.1875
20x15	1146	525	2770	6	3600	462	.2250

Steam Cylinder Lubrication

The steam cylinder of the compressor requires a good grade of steam cylinder oil, and due to its constant washing away by the steam, it should be fed to the cylinder in greater quantities than in the air cylinder.

The following table gives the approximate quantity of oil for steam cylinder lubrication.

TABLE V. OIL REQUIRED FOR STEAM-CYLINDER LUBRICATION.

Size of Cylinder, Inches	Drops of Oil per Minute	Pints of Oil per 10 Hours
6⅛x 6	3	.3
8 x 8	4	.4
10 x10	6	.6
12 x12	8	.8
14 x15	11	1.1
16 x10	11	1.1
18 x15	14	1.4
20 x12	14	1.4
24 x15	20	2 0

Inspection and Cleaning of Air Compressors

At stated intervals, say once a month, the compressor should be carefully inspected, lost motion in bearings taken up, and any defect corrected.

The best of lubricating oils will cause a deposit of carbon in the compressor system, and for this reason the air cylinder should be cleaned occasionally probably once a month, by filling the lubricator with a strong solution of water and soap, and feeding liberally during a day's run. Careful attention to this will avoid an accumulation of carbon in the cylinder and pipe line.

The oil in the crank case should be drawn out once a month, and the frame thoroughly washed out with kerosene, and then wiped out clean with a cloth. The oil may be used over again if it is properly filtered.

Circulating Water

The duty of the jacket water is to carry off the heat of compression, and to do this successfully requires that the supply of cooling water be liberal. The air cylinders are provided with water inlet openings at the bottom, and the outlet at the top. The water outlet should be in plain view of the operator to insure that the water is circulating, and it is best to arrange this by allowing the water to flow into an open funnel.

General Construction of Air Compressors.

In the modern air compressor, the frames are of the enclosed type, and are of either the center crank or side crank design. The center crank design illustrated in Fig. 9 is used on single compressors. The side crank design illustrated in Fig. 10 is of the rolling mill type, and is used only on duplex compressors.

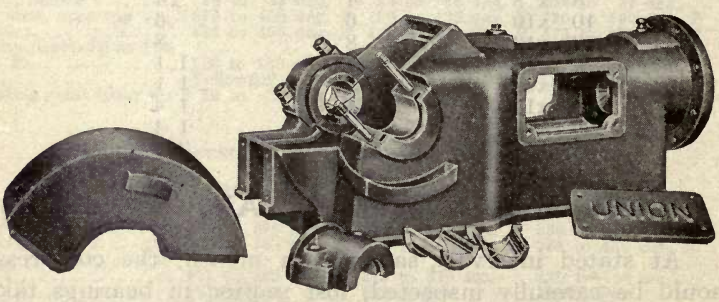


Fig. 9.

The frames are of massive proportion with metal properly distributed to secure the maximum strength and rigidity. They are extended to the foundation in the form of a broad, liberal support reaching their entire length. The crosshead guides are bored at the same setting as the fittings for the cylinders, thus insuring perfect alignment.

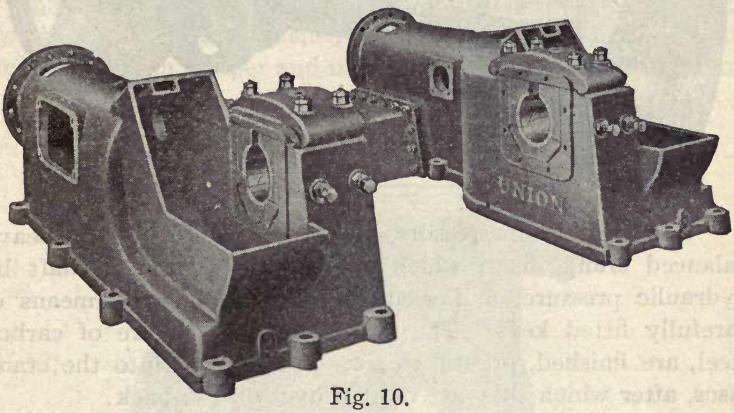


Fig. 10.

All bearings are lubricated by the splash system from the main frame. Close fitting cast iron covers prevent any leakage from the frame.

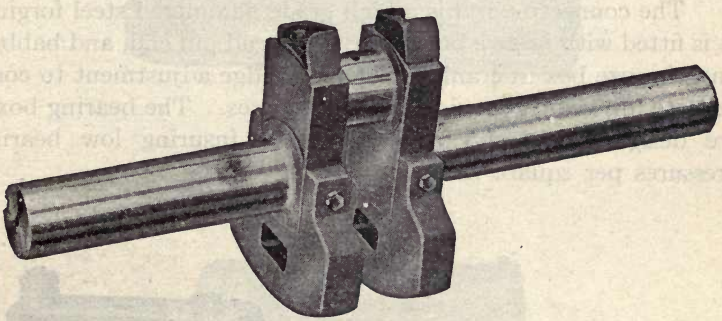


Fig. 11.

Crank Shaft.

The crank shaft is made of high carbon steel, and is accurately finished and ground to size.

On single compressors and small duplex compressors,

the shaft is of the center crank type with counter weights securely fastened to each crank.

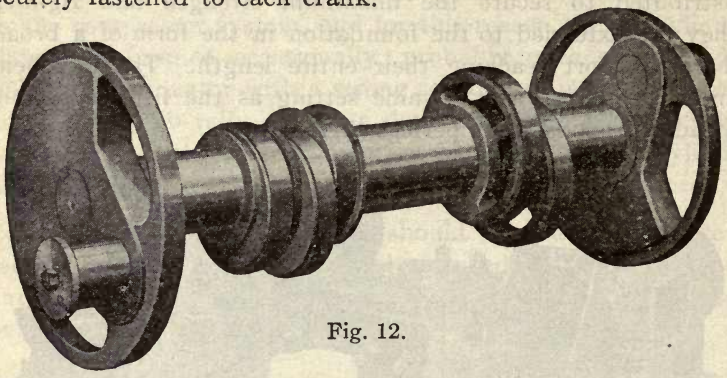


Fig. 12.

On duplex compressors, the shaft is fitted with heavy balanced crank discs, which are forced on to the shaft by hydraulic pressure, and securely held in place by means of carefully fitted keys. The crank pins, which are of carbon steel, are finished, ground to size, and pressed into the crank discs, after which they are riveted over on the back.

Connecting Rod

The connecting rod is a high grade hammered steel forging. It is fitted with bronze box in the crosshead pin end, and babbitt lined bronze box in crank pin end. Wedge adjustment to compensate for wear is provided for both boxes. The bearing boxes are designed with liberal proportions, insuring low bearing pressures per square inch.

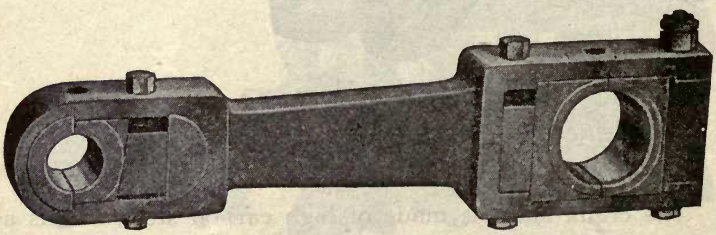


Fig. 13.

Crosshead

The crosshead is a steel casting and is provided with adjustable parallel-fitted bronze shoes top and bottom. It has unusual large bearing surface and the bearing pressure per square inch is correspondingly low.

Crosshead pin is made of best grade carbon steel and is accurately finished, hardened and ground. It is fitted into the crosshead on a taper and is secured by a nut and a lock nut.

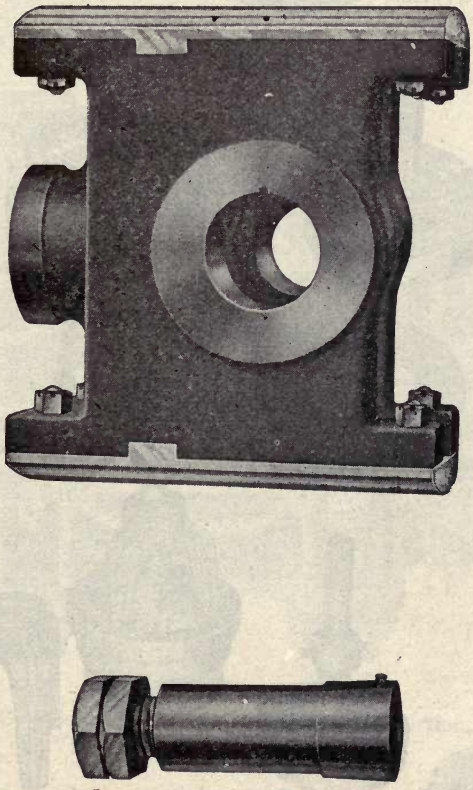


Fig. 14.

Air Valves

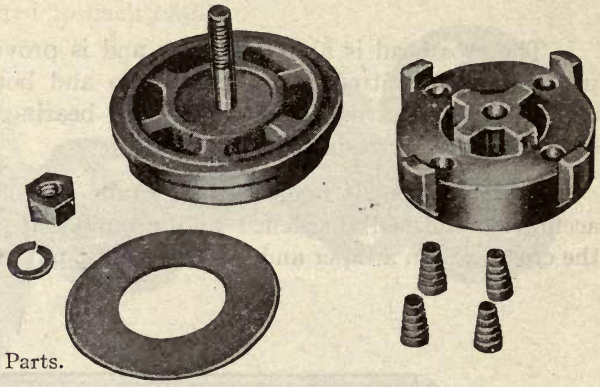


Fig. 226.
Discharge Valve Parts.

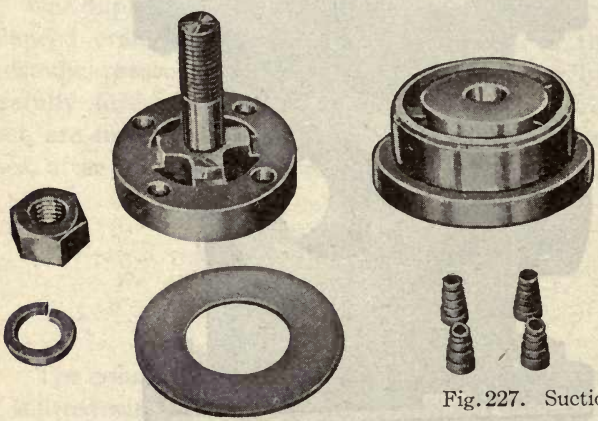


Fig. 227. Suction Valve Parts.

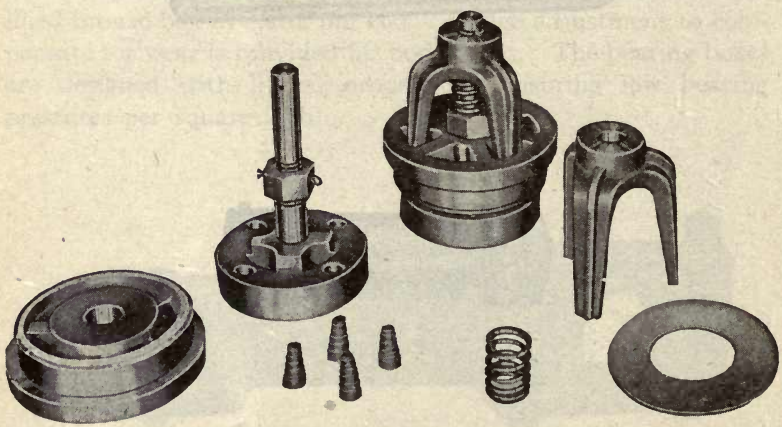


Fig. 228.
Suction Valve Parts Used on Class "BL" Air Compressors.

The valves and their arrangement are the most vital parts in the construction of an air compressor. No pains have been spared to make these parts of the best quality and design. Both suction and discharge valves are of the flat-disc type and are made of special grade, heat-treated steel, ground to size. Suction and discharge valves are of the same size and interchangeable. These valves which with their seats and guards constitute single units, are easily accessible by the removal of screwed caps. The valve-seats are of the double-ported type, which arrangement gives a maximum opening with very slight lift of valve. These features practically eliminate all noise of operation and insure a long life of the valve at high speeds.

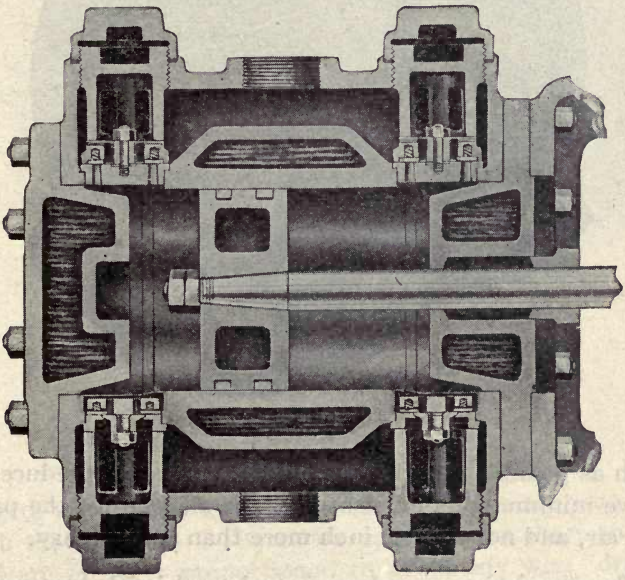


Fig. 19.
Air-Cylinder Section, Showing Piston Construction, Valves, Water-Jacketing Spaces.

Air-Cylinders.

The air-cylinders are made of semi-steel, counterbored at the ends, and have water jackets extending entirely around cylinders. Both heads are also water-jacketed, and special attention has been paid to the circulation of the cooling water, in order to get the maximum cooling effect with the least amount of water. With the object of making the volumetric efficiency

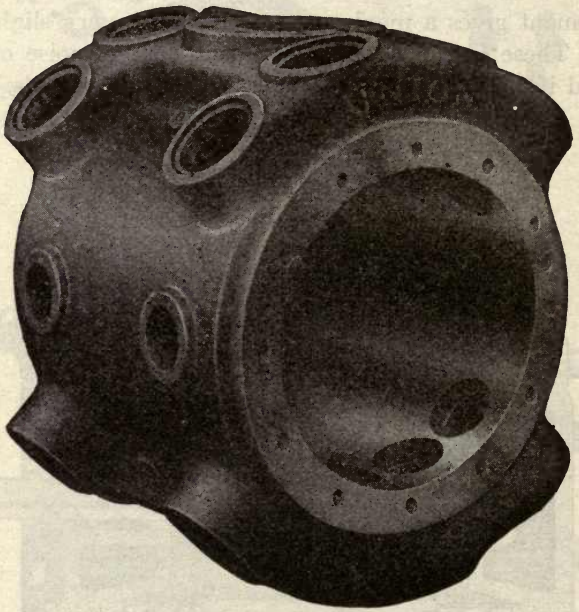


Fig. 20.

as high as possible, the clearance spaces have been reduced to a positive minimum. Sufficient space is allowed for the passage of the air, and not a cubic inch more than is necessary.

The air-pistons are of semi-steel, each with two carefully fitted snap rings. Pistons are fastened to piston rods by taper and nuts.

Valve Gear.

On steam-driven compressors, the steam valves are operated by eccentrics on the crank-shaft. These eccentrics are lubricated from the crank-case, thus avoiding the use of any grease or oil cups. The valve-gears with the fewest num-

ber of working parts possible, are coupled direct, avoiding any offsets or the use of rock arms, which may give trouble. Take-up for wear is provided at every bearing.

Steam Cylinders.

The steam cylinders which are made of semi-steel, are amply heavy to permit of reboring. They are counterbored at the ends to prevent the rings wearing a shoulder, and are

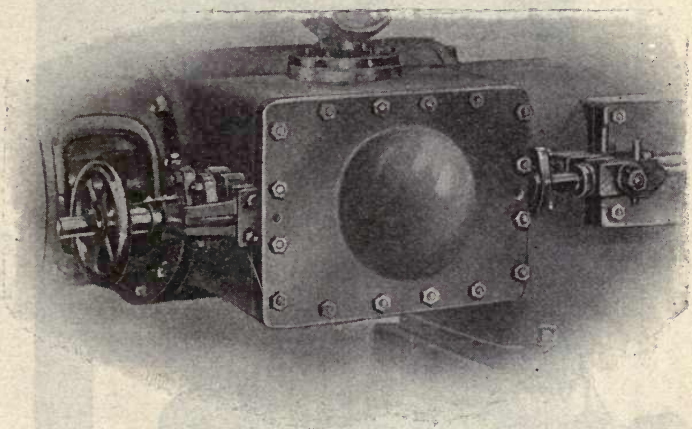


Fig. 21.

lagged with black iron. The steam chests are cast integral with the cylinders. The steam openings are short and direct, and are properly proportioned to avoid any wire drawing of steam, and reduce clearance to a minimum

The steam-cylinders are equipped with flat-faced slide valves, and cylinders 10 x 10 and larger have balancing pistons to balance the steam pressure on the back of the main valve to reduce the friction and wear to a minimum. Compressors with 12 x 12 steam-cylinders and larger have the Meyer cut-off adjustable to give cut-off from one-quarter to three-quarter stroke.

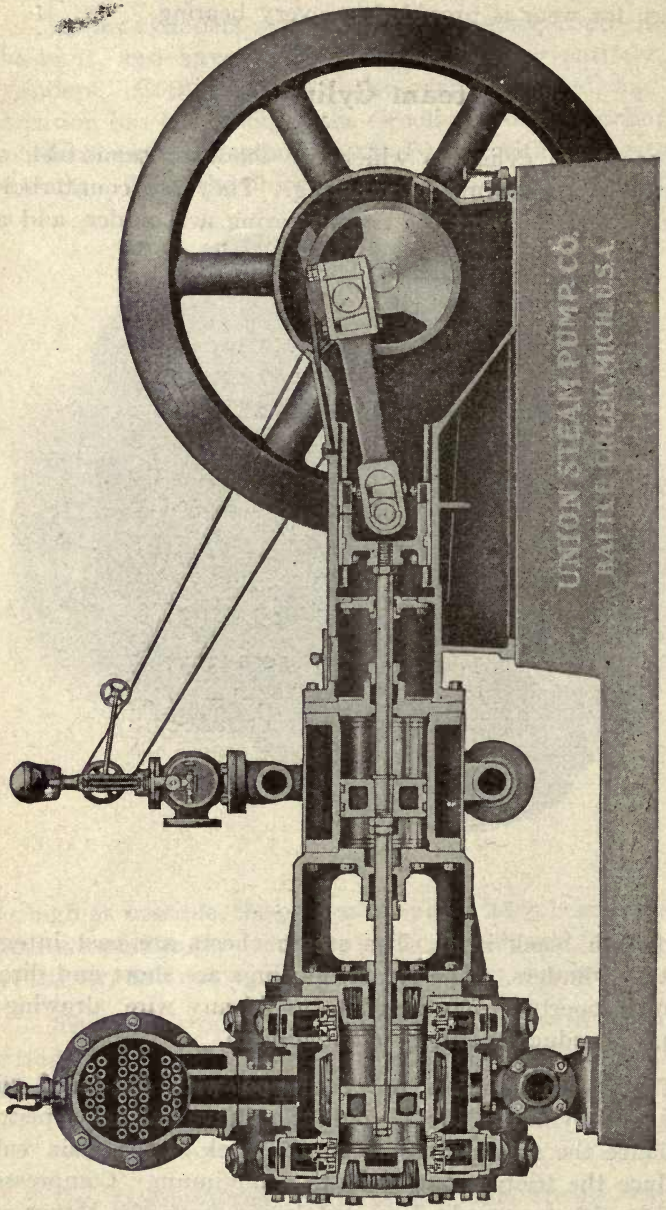


Fig. 22.
Longitudinal Section of Union Class "STD" Duplex Steam-Driven Two-Stage Enclosed-Type Air Compressor.

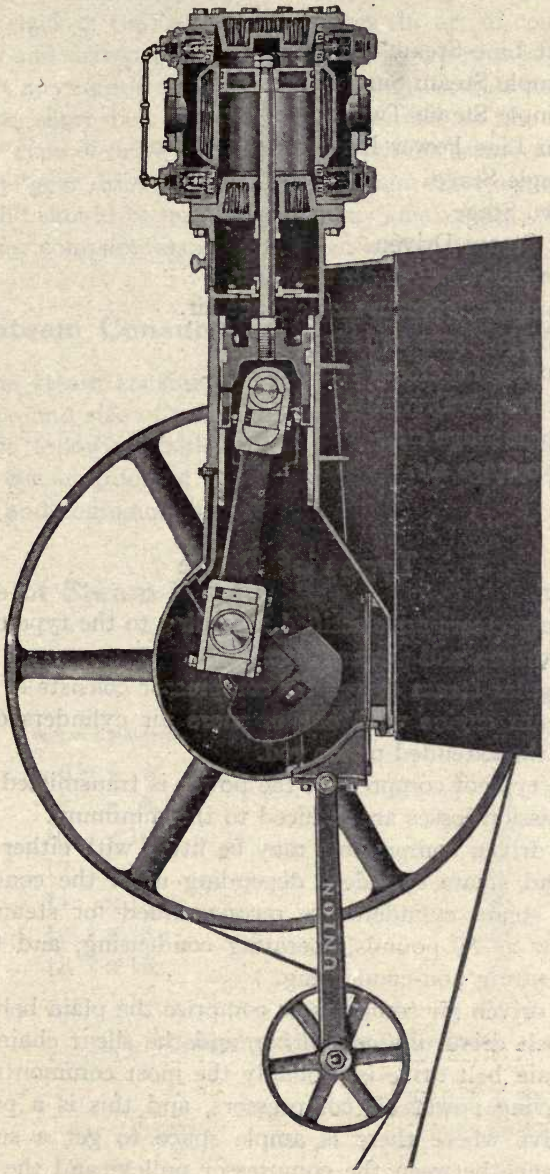


Fig. 229.
Longitudinal Section of Class "BL" Belt-Driven Compressor.

Classification of Air Compressors

Air compressors are classified according to their design as follows:

Straight Line Steam Driven.
- Simple Steam Single Stage Air.
- Simple Steam Two Stage Air.

Straight Line Power Driven.
- Single Stage.
- Two Stage.

Duplex Steam Driven.
- Simple steam single stage air.
- Compound steam single stage air.
- Simple steam two stage air.
- Compound steam two stage air.

Duplex Power Driven.
- Single stage air.
- Two stage air.

Types of Drive

Air Compressors are classified according to the type of drive into steam driven and power driven.

The regular steam driven air compressor consists of a very efficient steam engine with one or more air cylinders directly coupled on the extended piston rod.

In this type of compressor, the power is transmitted direct, and transmission losses are reduced to the minimum.

Steam driven compressors may be fitted with either simple or compound steam cylinders, depending upon the conditions. Compound steam cylinders are recommended for steam pressures as low as 80 pounds, operating condensing, and for 100 pounds operating non-condensing.

Power driven air compressors comprise the plain belt drive, the short belt drive, the gear drive, and the silent chain drive.

The plain belt drive is probably the most common method used in driving power air compressors, and this is a practical type of drive where there is ample space to get a sufficient center distance between the compressor pulley, and the motor. This center distance should be three to four times the diameter of the compressor pulley, and the direction of the belt motion should be such as to put the slack on top.

Where ample belt centers cannot be obtained, the short belt drive is advisable. This arrangement consists of a floating idler pulley and a very short belt. The weight of the idler takes up the slack of the belt, and increases the arc of contact on the driving and driven pulleys, so that the full power is transmitted without any undue strain on the bearings, or belt itself.

The short belt drive is recommended as the most satisfactory type of drive for a power driven air compressor.

The gear drive, and the silent chain drive are never advisable, and should be used only on very small units where the demand for compactness renders them imperative.

Steam Consumption of Air Compressors

The steam consumption of an air compressor varies with the type and size of machine, and the conditions of operation.

The following table will give an idea of the approximate steam consumption of air compressors of different sizes with simple and compound steam cylinders.

Table of Steam Consumption of Air Compressors

Simple Steam Cylinders.

Size of Cylinders	Steam per I. H. P. per hour, non-condensing, 100-125 pounds steam pressure.
6⅛ x 6	46
8 x 8	42
10 x 10	40
12 x 12	33
14 x 15	30
16 x 15	29
18 x 15	28

Compound Steam Cylinders.

Size of Cylinders	Steam per I. H. P. per hour, non-condensing, 100-125 pounds steam pressure.
8 and 12 x 8	30
10 and 16 x 10	28
12 and 20 x 12	26
14 and 24 x 15	25

Indicated or Brake Horse Power of an Air Compressor

The method of calculating the theoretical horse power to compress air by single stage and two stage compression has already been shown, and on pages 66 and 67, the theoretical horse power for various pressures in one, two, three and four stage compression is tabulated.

The theoretical figures given do not take into consideration the losses in the air cylinders due to clearance, the heat of compression, etc., nor the mechanical losses in the operation of the compressor. Consequently, to arrive at the indicated or brake horse power to compress a cubic foot of free air, it is necessary to take the indicated horse power of a steam driven compressor, and the brake horse power of a power driven compressor, and measure the actual free air delivered corrected to the suction temperature and pressure.

The indicated or brake horse power to compress a cubic foot of free air may then be accurately determined, and all losses are taken into consideration.

Due to these varying conditions in the compression of air, it is a more convenient method of calculation to base the indicated or brake horse power on the displacement of the compressor.

The following tables give the approximate indicated or brake horse power to compress air by single or two-stage compression per cubic foot of compressor displacement. In comparing the horse power figures for single and two-stage compression, it should be borne in mind that a two-stage compressor has a higher volumetric efficiency than a single-stage machine.

Approximate Indicated or Brake Horse Power to Compress 1 Cubic Foot of Compressor Displacement per Minute by Single Stage Compression at Sea Level.

Gauge Pressure	Horse Power	Gauge Pressure	Horse Power	Gauge Pressure	Horse Power
5	.0235	45	.124	85	.160
10	.0435	50	.130	90	.167
15	.062	55	.132	95	.172
20	.0756	60	.135	100	.178
25	.090	65	.143	105	.182
30	.102	70	.150	110	.190
35	.112	75	.155	125	.206
40	.12	80	.158		

Approximate Indicated or Brake Horse Power to Compress 1 Cubic Foot of Compressor Displacement per Minute by Two Stage Compression at Sea Level.

Gauge Pressure	Horse Power	Gauge Pressure	Horse Power
80	.157	300	.288
90	.165	350	.305
100	.177	400	.319
125	.189	450	.332
150	.210	500	.345
175	.221		
200	.242		
250	.265		

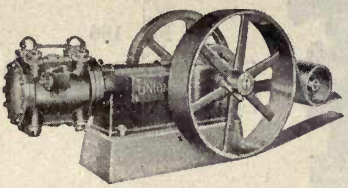


Fig. 24. Class "BL" Belt-Driven Air Compressor, Enclosed Type.

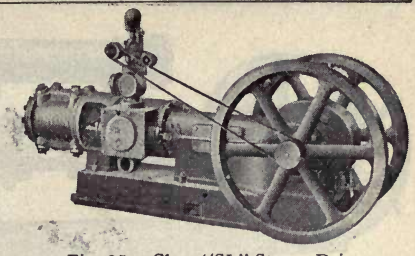


Fig. 25. Class "SL" Steam-Driven Air Compressor, Enclosed Type.

Class "BL" Belt-Driven Air Compressors, Enclosed Type.

Air Cylinder		Maximum Speed R. P. M.	Displacement cu. ft. Free Air		Maximum Air Pressure	Brake Horse-Power Required at Pressure Stated	Pipe Openings			Driving Pulley	
Diameter	Stroke		Per Revolution	Per Minute			Suction	Discharge	Cooling Water Inlet	Diameter	Width of Belt
6⅛	6	350	.205	72	125	11½	2	2	¼	28	6
7	6	350	.267	93	100	14	2½	2½	⅜	28	6
8	6	350	.349	122	50	15	2½	2½	⅜	28	6
8	8	300	.467	140	125	23	2½	2½	⅜	42	8
9	8	300	.590	177	100	26	3	3	⅜	42	8
10	8	300	.730	219	75	31	3	3	⅜	42	8
12	8	300	1.05	315	50	39	3½	3½	½	42	8
10	10	275	.91	250	125	47	3	3	½	48	10
12	10	275	1.30	358	100	60	4	4	½	54	10
14	10	275	1.78	490	60	65	4½	4½	½	54	10
10	12	250	1.09	273	125	50	3½	3½	½	54	10
12	12	250	1.58	395	125	74	4	4	½	63	12
14	12	250	2.14	535	100	87	4½	4½	½	63	12
16	12	250	2.79	698	60	94	5	5	½	63	12
18	12	250	3.54	885	40	100	6	6	½	66	12
14	15	220	2.68	590	125	110	4½	4½	½	72	15
16	15	220	3.49	768	100	125	5	5	½	72	15
18	15	220	4.42	975	60	132	6	6	½	72	15
20	15	220	5.46	1200	40	140	7	7	½	72	15

Class "SL" Steam-Driven Air Compressors, Enclosed Type.

Size of Compressor			Maximum Speed R. P. M.	Displacement cu. ft. Free Air		Maximum Air Pressure	Required Steam Pressure ¾ Cut Off	I. H. P. of Steam Cyl. at Pressure Stated	Pipe Openings				
Diameter Steam Cylinder	Diameter Air Cylinder	Stroke		Per Revolution	Per Minute				Steam	Exhaust	Suction	Discharge	Cooling Water Inlet
6⅛	6⅛	6	350	.205	72	125	90	11½	1½	2	2	2	¼
6⅛	7	6	350	.267	93	100	100	14	1½	2	2½	2½	⅜
6⅛	8	6	350	.349	122	50	120	15	1½	2	2½	2½	⅜
8	8	8	300	.467	140	125	90	23	2	2½	2½	2½	⅜
8	9	8	300	.59	177	100	110	26	2	2½	3	3	⅜
8	10	8	300	.73	219	75	100	31	2	2½	3	3	⅜
8	12	8	300	1.05	315	50	110	39	2	2½	3½	3½	½
10	10	10	275	.91	250	125	90	47	3	3½	3	3	½
10	12	10	275	1.30	358	100	125	60	3	3½	4	4	½
10	14	10	275	1.78	490	60	115	65	3	3½	4½	4½	½
10	10	12	250	1.09	273	125	90	50	3	3½	3½	3½	½
10	12	12	250	1.58	395	125	125	74	3	3½	4	4	½
10	14	12	250	2.14	535	100	150	87	3	3½	4½	4½	½
12	10	12	250	1.09	273	125	75	50	3½	4	3½	3½	½
12	12	12	250	1.58	395	125	90	74	3½	4	4	4	½
12	14	12	250	2.14	535	100	115	87	3½	4	4½	4½	½
12	16	12	250	2.79	698	60	110	94	3½	4	5	5	½
12	18	12	250	3.54	885	40	110	100	3½	4	6	6	½
14	14	15	220	2.68	590	125	90	110	4	5	4½	4½	½
14	16	15	220	3.49	768	100	105	125	4	5	5	5	½
14	18	15	220	4.42	975	60	100	132	4	5	6	6	½
14	20	15	220	5.46	1200	40	100	140	4	5	7	7	½

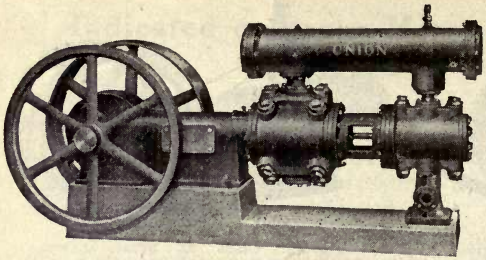


Fig. 186

Class "TTBL" Belt-Driven Air Compressor, Double-Acting Two-Stage, Enclosed Type.

Cylinders			Maximum Speed R. P. M.	Displacement Cu. Ft. Free Air		Maximum Air Pressure	Brake Horsepower at 100 Lbs. Pressure	Pipe Openings		Driving Pulley	
Diameter of Low Pressure Air-Cyl.	Diameter of High Pressure Air-Cyl.	Stroke		Per Revolution	Per Minute			Suction	Discharge	Diameter	Width of Belt
8	5	6	350	.34	122	125	22	2½	2	28	6
10	6⅛	8	300	.73	219	150	45	3	2½	42	8
12	7	8	300	1.05	315	125	58	3½	2½	42	8
12	6⅛	10	275	1.30	358	200	85	4	2½	54	10
12	7	10	275	1.30	358	125	65	4	2½	54	10
14	8	10	275	1.78	490	125	90	4½	2½	54	10
14	8	12	250	2.14	535	200	130	4½	3	66	12
16	9	12	250	2.79	698	125	130	5	3½	66	12
18	10	15	220	4.42	975	125	180	6	4½	72	15
20	12	15	220	5.46	1200	100	210	7	4	72	15

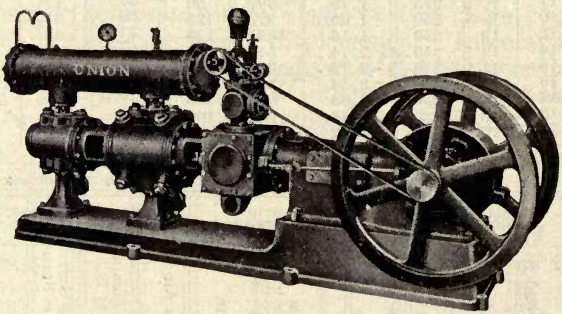


Fig. 187

Class "TTSL" Steam-Driven Double-Acting Two-Stage Compressor, Enclosed Type.

Cylinders				Maximum Speed R. P. M.	Displacement Cu. Ft. Free Air		Maximum Air Pressure	Required Steam Pressure	I. H. P. of Steam Cyl. at 100 Lbs. Air Press.	Pipe Openings			
Diameter of Steam-Cylinder	Diameter of Low Pressure Air-Cyl.	Diameter of High Pressure Air-Cyl.	Stroke		Per Revolution	Per Minute				Steam	Exhaust	Suction	Discharge
8	8	5	6	350	.349	122	125	80	22	2	2½	2½	2
10	10	6⅛	8	300	.73	219	1 0	90	45	2½	3	3	2½
10	12	7	8	300	1.05	315	125	110	58	2½	3	3½	2½
12	12	6⅛	10	275	1.30	358	200	100	85	3½	4	4	2½
12	12	7	10	275	1.30	358	125	85	65	3½	4	4	2½
12	14	8	10	275	1.78	490	125	100	90	3½	4	4½	2½
14	14	8	12	250	2.14	535	200	100	130	4	5	4½	3
14	16	9	12	250	2.79	698	125	100	130	4	5	5	3½
18	18	10	15	220	4.42	975	125	90	180	5	6	6	4½
18	20	12	15	220	5.46	1200	100	100	210	5	6	7	4½

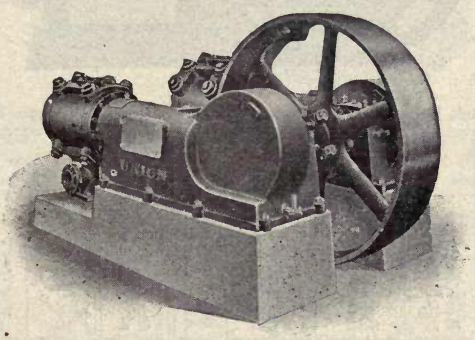


Fig 188

High Pressure Duplex Belt Driven Gas Compressors
For Extraction of Gasoline From Natural Gas

Size of Compressor			CAPACITY					Driving Pulley		Size of Openings For long pipe lines, use larger pipes, reducing size at compressor		
Diam. of Low-Pressure Cyl.	Diam. of High-Pressure Cyl.	Stroke	Revolutions per Minute	Displacement Cu. Feet. Free Gas Per Minute	Displacement Cu. Ft. Free Gas Per 24 Hrs.	Maximum Air-Pressure	Brake H. Power At Stated Pressure	Diameter	Width of Belt	Suction	Discharge	Cooling Water Inlet
6 1/8	3	6	300	61	88,560	300	17	42	8	2	2	1/2
8	4	6	300	104	150,760	300	29	42	8	2 1/2	2	1/2
10	5	10	250	227	327,600	300	64	54	14	3	2	3/4
12	6	12	235	371	532,940	300	100	66	18	4	2 1/2	1
14	7	12	235	501	724,175	300	135	66	18	4 1/2	3	1
16	8	15	210	730	1,055,370	300	200	80	25	5	3 1/2	1
18	9	15	210	929	1,337,600	250	250	80	25	6	4	1

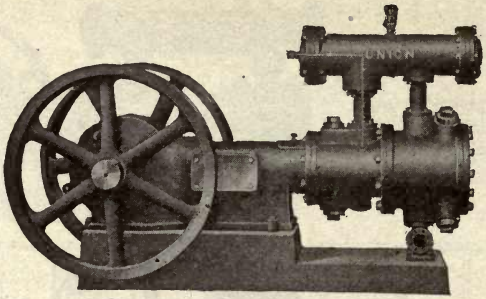


Fig. 26

Class "BTL" Belt-Driven Air Compressor, Single-Acting, Two-Stage, Enclosed Type.

Cylinders			Revolutions per Minute	Displacement cu. ft. Free Air		Maximum Air Pressure	Brake Horsepower at Pressure Stated	Pipe Openings		Driving Pulley	
Diameter of Low Pressure Air-Cyl.	Diameter of High Pressure Air-Cyl.	Stroke		Per Revolution	Per Minute			Suction	Discharge	Diameter	Width of Belt
8	4	6	275	.173	47.5	250	12.5	2 ½	1 ½	28	6
10	5	6	275	.262	72	200	17.5	3	1 ½	28	6
10	5	8	265	.364	96	250	25	3	1 ½	42	8
12	6 ⅛	8	265	.524	139	250	37	3 ½	2	42	8
12	6 ⅛	10	250	.655	164	250	44	4	2 ½	48	10
14	7	10	250	.89	223	250	60	4 ½	2 ½	54	10
16	8	12	235	1.4	328	250	88	5	3	66	12

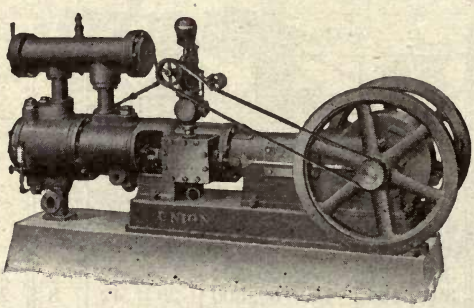


Fig. 27

Class "STL" Steam-Driven, Single-Acting, Two-Stage Compressor, Enclosed Type.

Cylinders				Revolutions per Minute	Displacement cu. ft. Free Air		Maximum Air Pressure	Required Steam Pressure	I. H. P. of Steam Cyl. at Pressure Stated	Pipe Openings			
Diameter of Steam-cylinder	Diameter of Low Pressure Air-Cyl.	Diameter of High Pressure Air-Cyl.	Stroke		Per Revolution	Per Minute				Steam	Exhaust	Suction	Discharge
6 ⅛	8	4	6	275	.173	47.5	250	110	12.5	1 ½	2	2 ½	1 ½
6 ⅛	10	5	6	275	.262	72	200	150	17.5	1 ½	2	3	1 ½
8	10	5	8	265	.364	96	250	100	25	2	2 ½	3	1 ½
8	12	6 ⅛	8	265	.524	139	250	110	37	2	2 ½	3 ½	2
10	12	6 ⅛	10	250	.655	164	250	100	44	3	3 ½	4	2 ½
10	14	7	10	250	.89	223	250	115	60	3	3 ½	4 ½	2 ½
12	16	8	12	235	1.4	328	250	100	88	3 ½	4	5	3

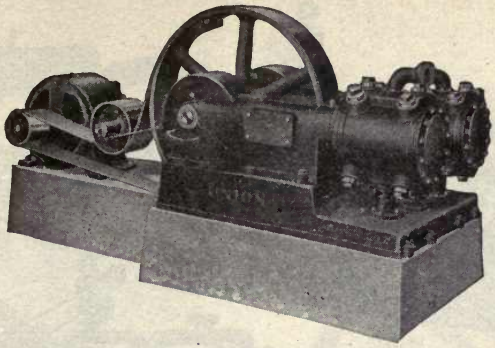


Fig. 28.

Class "DBL" Duplex, Short-Belt Driven Center Crank Air Compressor, Enclosed Type.

Size of Compressor		Capacity					Driving Pulley		Size of Openings		
			Displacement cubic feet Free Air						For long pipe lines, use larger pipes, reducing size at compressor		
Diameter Air Cylinders	Stroke	Revolutions per Minute	Per Revolution	Per Minute	Maximum Air Pressure	Brake Horse-Power at Pressure Stated	Diameter	Width of Belt	Suction	Discharge	Cooling Water Inlet
6 1/8	6	275	.41	112	100	20	42	8	2 1/2	2 1/2	1/2
8	6	275	.70	192	50	25	42	8	3	3	1/2
8	8	265	.93	246	100	44	54	10	3	3	3/4
9	8	265	1.18	312	85	50	54	10	3 1/2	3 1/2	3/4
10	8	265	1.46	386	75	60	54	10	3 1/2	3 1/2	3/4
12	8	265	2.10	556	50	72	54	10	4 1/2	4 1/2	3/4

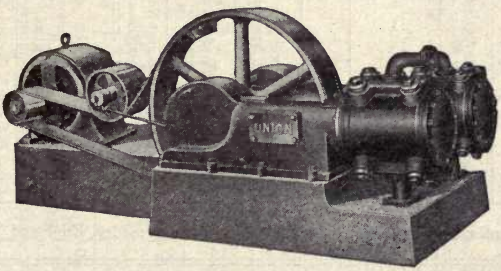


Fig. 29.

Class "BD" Duplex Short-Belt-Driven Side Crank, Air Compressor, Enclosed Type.

Size of Compressor		Capacity					Driving Pulley		Size of Openings		
			Displacement Cubic Feet Free Air						For long pipe lines, use larger pipes, reducing size at compressor		
Diameter Air Cylinders	Stroke	Revolutions per Minute	Per Revolution	Per Minute	Maximum Air Pressure	Brake Horse-Power at Pressure Stated	Diameter	Width of Belt	Suction	Discharge	Cooling Water Inlet
10	10	250	1.82	456	80	72	54	14	4	4	3/4
12	10	250	2.60	650	60	88	54	14	5	5	3/4
14	10	250	3.56	890	40	107	54	14	6	6	3/4
10	12	235	2.18	512	100	92	66	18	4	4	1
12	12	235	3.14	740	100	132	66	18	5	5	1
14	12	235	4.28	1002	70	150	66	18	6	6	1
16	12	235	5.58	1310	50	170	66	18	7	7	1
18	12	235	7.08	1660	40	200	66	18	8	8	1
14	15	210	5.36	1122	90	188	80	25	6	6	1
16	15	210	6.98	1460	70	220	80	25	7	7	1
18	15	210	8.84	1858	50	240	80	25	8	8	1
20	15	210	10.92	2292	40	275	80	25	8	8	1

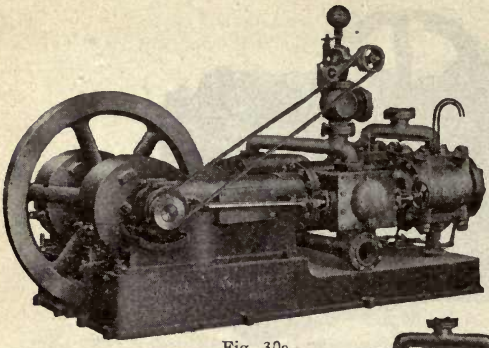


Fig. 30a

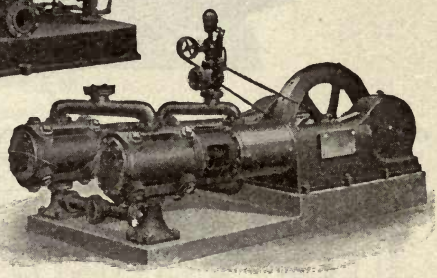


Fig. 30b

Class "DSL" Duplex Steam-Driven Center Crank Air Compressors, Enclosed Type. Fig. 30a

Size of Compressor			Capacity						Size of Openings For long pipe lines, use larger pipes, reducing size at compressor				
Diameter Steam Cyls.	Diameter Air Cylinders	Stroke	Revolutions per Minute	Displacement Cubic Feet Free Air: Per Revolution	Displacement Cubic Feet Free Air: Per Minute	Maximum Air Pressure	Required Steam Pressure	Indicated Horse-Power at Air Pressure Stated	Steam	Exhaust	Suction	Discharge	Cooling Water Inlet
6	6 1/8	6	275	.41	114	100	75	20	2	2 1/2	2 1/2	2 1/2	1/2
6	8	6	275	.693	192	50	120	25	2	2 1/2	3	3	1/2
8	8	8	265	.934	248	100	75	44	3	3 1/2	3	3	3/4
8	9	8	265	1.18	312	85	95	50	3	3 1/2	3 1/2	3 1/2	3/4
8	10	8	265	1.46	386	75	100	60	3	3 1/2	3 1/2	3 1/2	3/4
8	12	8	265	2.10	556	50	110	72	3	3 1/2	4 1/2	4 1/2	3/4

Class "SD" Duplex Steam-Driven Side Crank Air Compressors, Enclosed Type. Fig. 30b

Size of Compressor			CAPACITY						Size of Openings For long pipe lines, use larger pipes, reducing size at compressor				
Diameter Steam Cyls.	Diameter Air Cylinders	Stroke	Revolutions per Minute	Displacement Cubic Feet Free Air: Per Revolution	Displacement Cubic Feet Free Air: Per Minute	Maximum Air Pressure	Required Steam Pressure	Indicated Horse-Power at Air Pressure Stated	Steam	Exhaust	Suction	Discharge	Cooling Water Inlet
10	10	10	250	1.82	456	80	70	72	4	4 1/2	4	4	3/4
10	12	10	250	2.60	650	60	90	88	4	4 1/2	5	5	3/4
10	14	10	250	3.56	890	40	100	107	4	4 1/2	6	6	3/4
10	10	12	235	2.18	512	100	75	92	4	4 1/2	4	4	1
10	12	12	235	3.16	740	100	110	132	4	4 1/2	5	5	1
10	14	12	235	4.28	1002	70	120	150	4	4 1/2	6	6	1
12	12	12	235	3.16	742	100	75	132	5	6	5	5	1
12	14	12	235	4.28	1002	70	90	150	5	6	6	6	1
12	16	12	235	5.58	1310	50	100	170	5	6	7	7	1
12	18	12	235	7.18	1660	40	110	198	5	6	8	8	1
14	14	15	210	5.36	1122	90	75	188	6	7	6	6	1
14	16	15	210	6.98	1460	70	90	220	6	7	7	7	1
14	18	15	210	8.84	1858	50	90	240	6	7	8	8	1
14	20	15	210	10.92	2292	40	100	275	6	7	8	8	1

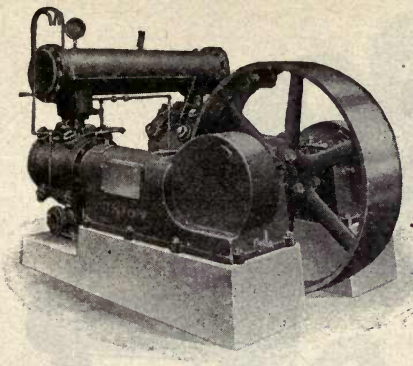


Fig. 32

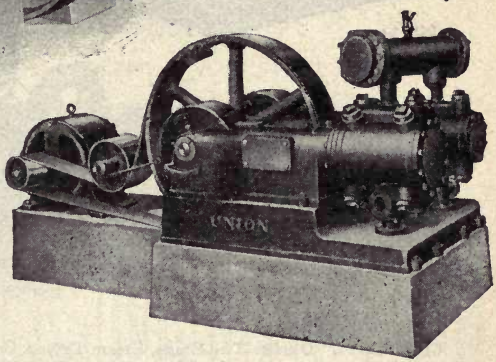


Fig. 31

Class "DBTL" Duplex Two-Stage, Short-Belt Driven, Center Crank Air Compressor, Enclosed Type. **Fig. 31.**

Size of Compressor			CAPACITY					Driving Pulley		Size of Opening		
				Displacement Cubic Feet Free Air						For long pipe lines, use larger pipes, reducing size at Compressor		
Diameter of Low-Pressure Air-Cylinder	Diameter of High-Pressure Air Cylinder	Stroke	Revolutions per Minute	Per Revolution	Per Minute	Maximum Air-Pressure	Brake Horse-Power at 100 Pounds Pressure	Diameter	Width of Belt	Suction	Discharge	Cooling Water Inlet
12	7	8	265	1.050	278	150	46	54	10	3½	2½	¾
12	8	8	265	1.050	278	125	46	54	10	3½	2½	¾

Class "BDT" Duplex Two-Stage, Short Belt-Driven, Side Crank Air Compressor, Enclosed Type. **Fig. 32.**

Size of Compressor			CAPACITY					Driving Pulley		Size of Openings		
				Displacement Cu. ft. Free Air.						For long pipe lines, use larger pipes, reducing size at compressor		
Diam. of Low-Pressure Air-Cyl.	Diam. of High-Pressure Air-Cyl.	Stroke	Revolutions per Minute	Per Revolution	Per Minute	Maximum Air-Pressure	Brake H. Power at 100 lbs. Pressure	Diameter	Width of Belt	Suction	Discharge	Cooling Water Inlet
12	7	10	250	1.310	325	150	54	54	14	4	2½	¾
12	8	10	250	1.310	325	125	54	54	14	4	2½	¾
14	7	10	250	1.780	445	150	74	54	14	4½	2½	¾
14	8	10	250	1.780	445	125	74	54	14	4½	2½	¾
16	10	12	235	2.800	655	150	110	66	18	5	3½	1
18	10	12	235	3.540	830	150	139	66	18	6	3½	1
18	12	12	235	3.540	830	100	139	66	18	6	4	1
18	12	15	210	4.420	929	125	155	80	25	6	4½	1
20	12	15	210	5.460	1146	125	190	80	25	7	4½	1

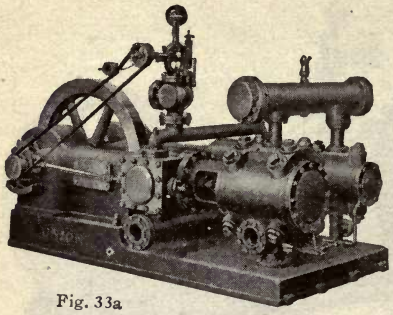


Fig. 33a

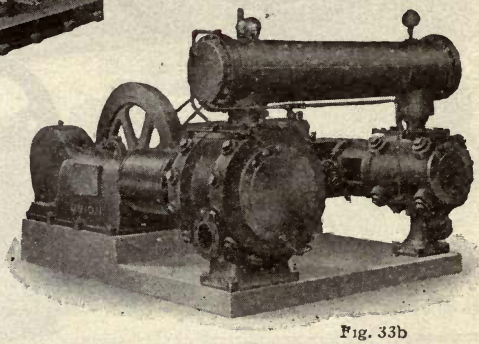


Fig. 33b

Class "DSTL" Duplex Two-Stage, Steam-Driven Center Crank Air Compressor, Enclosed Type Fig. 33a

SIZE OF COMPRESSOR				CAPACITY						Size of Openings				
					Displacem't Cubic Feet Free Air					For long pipe lines, use larger pipes, reducing size at compressor				
Diameter Steam Cylinders	Diameter Low-Pressure Air-Cylinder	Diameter High-Pressure Air-Cylinder	Stroke	Revolutions per Minute	Per Revolution	Per Minute	Maximum Air Pressure	Required Steam Pressure	Indicated Horse-Power at 100 Pounds Air Pressure	Steam	Exhaust	Suction	Discharge	Cooling Water Inlet
8	12	7	8	265	1.05	278	150	110	46	3	3 1/2	3 1/2	2 1/2	3/4
8	12	8	8	265	1.05	278	125	100	46	3	3 1/2	3 1/2	2 1/2	3/4

Class "SDT" Duplex Two-Stage, Steam-Driven Side Crank Air Compressor, Enclosed Type Fig. 33b

SIZE OF COMPRESSOR				CAPACITY						Size of Openings				
					Displacem't Cubic Feet Free Air					For long pipe lines, use larger pipes, reducing size at compressor				
Diameter Steam Cylinders	Diameter Low-Pressure Air-Cylinder	Diameter High-Pressure Air-Cylinder	Stroke	Revolutions Per Minute	Per Revolution	Per Minute	Maximum Air Pressure	Required Steam Pressure	Indicated Horse-Power at 100 Pounds Air Pressure	Steam	Exhaust	Suction	Discharge	Cooling Water Inlet
10	12	7	10	250	1.31	325	150	70	54	4	4 1/2	4	2 1/2	3/4
10	12	8	10	250	1.31	325	125	65	54	4	4 1/2	4	2 1/2	3/4
10	14	7	10	250	1.78	445	150	100	74	4	4 1/2	4 1/2	2 1/2	3/4
10	14	8	10	250	1.78	445	125	90	74	4	4 1/2	4 1/2	2 1/2	3/4
10	16	10	12	235	2.80	655	150	115	110	4	4 1/2	5	3 1/2	1
12	18	10	12	235	3.54	830	150	100	139	5	6	6	3 1/2	1
12	18	12	12	235	3.54	830	100	100	139	5	6	6	4	1
14	18	12	15	210	4.42	929	125	85	155	6	7	6	4 1/2	1
14	20	12	15	210	5.45	1146	125	100	190	6	7	7	4 1/2	1

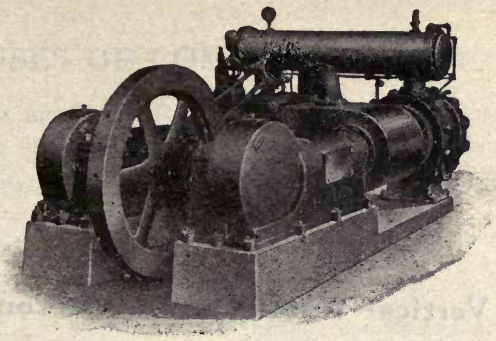


Fig. 34

Class "CTD" Duplex Cross Compound, Two-Stage, Side Crank, Air Compressor, Enclosed Type

SIZE OF COMPRESSOR					CAPACITY						SIZE OF OPENINGS				
						Displ'cm't Cubic Ft. Free Air					For long pipe lines, use larger pipes, reducing size at compressor				
Diam. of High Pressure Steam Cylinder	Diam. of Low Pressure Steam Cylinder	Diam. of Low Pressure Air Cylinder	Diam. of High Pressure Air Cylinder	Stroke	Revolutions per Minute	Per Revolution	Per Minute	Maximum Air Pressure	Required Steam Pressure	Indicated Horse-Power at 100 Pounds Air Pressure	Steam	Exhaust	Suction	Discharge	Cooling Water Inlet
10	16	12	7	10	250	1.31	325	150	90	54	3	5	4	2 ½	¾
10	16	12	8	10	250	1.31	325	125	80	54	3	5	4	2 ½	¾
10	16	14	7	10	250	1.78	445	150	115	74	3	5	4 ½	2 ½	¾
10	16	14	8	10	250	1.78	445	125	110	74	3	5	4 ½	2 ½	¾
12	20	16	10	12	235	2.79	656	150	100	110	3 ½	6	5	3 ½	1
12	20	18	10	12	235	3.54	831	150	110	139	3 ½	6	6	3 ½	1
12	20	18	12	12	235	3.54	831	100	100	139	3 ½	6	6	4	1
14	24	18	12	15	210	4.42	929	125	85	155	4	8	6	4 ½	1
14	24	20	12	15	210	5.45	1146	125	100	190	4	8	7	4 ½	1

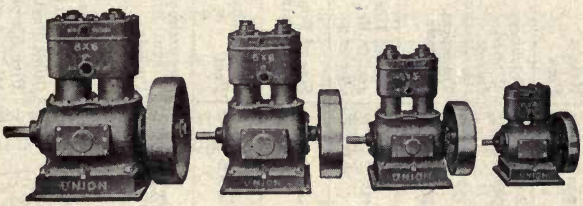


Fig. 35
Vertical Duplex High-Speed Air Compressors

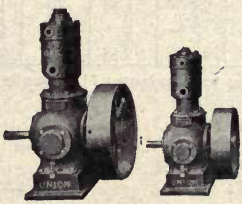


Fig. 36.
Vertical Single High-Speed Air Compressors.

Union Vertical Water-Cooled Air Compressors

	SIZE Bore and Stroke	SPEED R. P. M. Min.	SPEED R. P. M. Max.	Displacement cu. ft. Free Air per minute at Maximum Speed	Maximum Pressure Designed for	Brake H. P. required at 100 lbs. Pressure Maximum Speed	Openings		Fly-Wheel	
							Suction	Discharge	Diameter	Belt
Single	4½x5	300	450	21	200	4.25	1¼	1	18	4½
	6 x6	275	400	40	200	8.5	2	1½	24	5
Duplex	3½x4	350	550	25	200	5	1	¾	16	3
	4½x5	300	450	42	150	8.5	1¼	1	20	4½
	6 x6	275	400	80	125	17	2	1½	24	5
	8 x6	275	400	140	125	27	2½	2	28	6

Air Receivers

SIZE Diameter Inches	SIZE Height Feet	Thickness of Shell Inches	Thickness of Heads Inches	Size of Inlet and Outlet Openings	Size of Pop Safety-valve	Cubic Contents Feet	Approximate Weight Pounds	Suitable for Capacity of Cubic Feet Free Air per Minute of
12	3	5/16	¼	1 ¼	¾	2⅓	70	9 or less
18	3½	5/16	¼	1 ¼	¾	6	170	20 "
18	6	5/16	¼	1 ¼	1	10	300	90 "
20	5	5/16	¼	1 ½	1	10	300	100 "
24	5	5/16	¼	2	1	15	345	120 "
24	6	5/16	¼	2	1	18	435	130 "
24	8	5/16	¼	2	1	24	537	140 "
30	6	¼	⅜	3	1 ¼	30	730	150 "
36	6	¼	⅜	3 ½	1 ½	42	930	200 "
36	8	¼	⅜	4	1 ½	57	1200	300 "
42	8	¼	⅜	4	1 ½	77	1420	450 "
42	10	¼	⅜	5	2	96	1630	700 "
48	12	¼	⅜	6	2	150	2300	1100 "
54	12	5/16	5/16	6	2 ½	191	3000	1700 "

165 pounds hydraulic test pressure; 110 pounds working pressure.

USES OF COMPRESSED AIR

New uses are being found every day for compressed air. The following list shows a few of the many applications of compressed air, which will serve as a guide to those contemplating its use.

ACID WORKS
Agitating, Elevating and Transferring Acids and Acid Solutions
Pumping Water

ASPHALT REFINERIES
Agitating Asphalt
Scaling Asphalt Tanks or Vats

AGRICULTURAL IMPLEMENT MAKERS
Operating Pneumatic Hammers and Drills
Running Pneumatic Hoists and Lifts
Sand Blast for Cleaning Castings and Removing Paint
Pneumatic Painting and Whitewashing
Blowing Oil Furnaces
Cleaning Boiler Tubes

AUTOMOBILE GARAGES AND REPAIR SHOPS
Pumping up Tires
Cleaning Engines and Machines by the Air Jet
Operating Jacks, Lifts and Hoists
Running Pneumatic Hammers, Drills, Reamers, etc.
Cleaning Cars and Cushions with the Air Jet
Operating Brazing Forges
Supplying Oil Burners
Starting Gas and Gasoline Engines

BLEACHERIES
Pumping Water
Handling Chlorine or Bleaching Solutions
Agitating Liquids

BOILER SHOPS
Running Chipping, Riveting and Calking Hammers
Operating Drills, Reamers, Flue Rollers, Flue Expanders, Stay Bolt Cutters, Punches, etc.
Blowing Rivet Forges
Supplying Hoists, Lifts and Jacks
Cleaning Boiler Flues
Removing Scale, Rust and Paint by the Sand Blast

BREWERIES
Pumping Water
Racking Beer
Scaling Condenser Coils
Cleaning Boiler Tubes
Cleaning Machines by the Air Jet
Operating Air Hoists and Lifts
Refrigeration

BRIDGE BUILDERS
Operating Chipping and Riveting Hammers, Drills, Reamers, Punches, etc.
Supplying Air Hoists and Lifts
Blowing Rivet Forges
Cleaning Steel with the Sand Blast
Pneumatic Paint Spraying System

CEMENT BLOCK FACTORIES
Operating Sand Sifters
Running Concrete Rammers
Air Hoists and Lifts

CEMENT LAKES
Pumping Marl or Slurry by the Return-Air System

CEMENT MINES OR QUARRIES
Operating Rock Drills

CHEMICAL WORKS
Pumping Water
Agitating, Aerating, Elevating and Transferring Liquids
Testing and Calking Tanks and Pipe Lines

CHINA WORKS AND POTTERIES
Spraying Colors and Enamels

COAL MINES
Operating Coal Punchers, Chain Machines and Coal Cutters
Running Rock Drills and Hammer Drills
Pneumatic Haulage Systems
Return Air System for Station and Sump Pumping

Operating Direct-Acting Pumps
Running Coal Pick and Drill Sharpeners
Cleaning Boiler Flues
Pneumatic Tools for the Repair Shop
Air Hoists, Lifts and Motors
Pile Drivers for Shaft Sinking
Air Lift Pump for Water Supply or Mine Unwatering

CONTRACT WORK

Running Rock Drills, Hammer Drills and Stone Channelers
Operating Pneumatic Hammers, Drills and Reamers
Calking Pipe Lines and Tanks
Running Pumps
Plug Drills in Trenching
Sheet Pile Drivers
Drill Sharpeners
Blowing Smith Fires
Submarine Drills
Subaqueous Tunneling
Caisson Work
Steam Shovels

COTTON FINISHING WORKS

Pumping Water
Operating Baling Presses
Cleaning Presses, Slashers, and other Machines by the Air Jet
Agitating, Elevating and Transferring Dyes and Solutions
Air Hoists, Lifts and Motors
Automatic Sprinkler Systems
Humidifying Systems

COTTON MILLS

Pumping Water
Cleaning Looms, Lifts and Spindle Rails by the Air Jet
Humidifying Systems
Automatic Sprinkler Systems
Air Lifts, Hoists and Motors
Moistening Goods

COTTON OIL MILLS

Cleaning Crusher Rolls and Separator Plates
Air Hoists, Lifts and Motors
Pumping Water
Operating Formers

CREOSOTING PLANTS

Wood Preserving Processes

CUT STONE AND MONUMENTAL YARDS

Running Stone Tools, Polishers, etc.
Lettering and Carving
Cleaning, Carvings with the Air Jet
Air Hoists and Lifts
Running Plug Drills and Brush Hammers

DYE WORKS

Pumping Water
Agitating, Elevating and Transferring Dyes

ELECTRIC POWER AND LIGHTING PLANTS

Operating Air Hoists
Cleaning Engines and Generators with the Air Jet
Cleaning Boiler Flues
Calking Boilers

ELECTRIC RAILWAYS

Cleaning Motors and Generators with the Air Jet
Pneumatic Hammers and Drills
Air Hoists, Lifts and Jacks
Blowing Forges
Cleaning Cars by the Air Jet
Pumping Water
Air Brakes
Storage Air Brake Systems
Switch and Signal Systems
Operating Car Doors.
Track Sanders
Cleaning Rails by the Sand Blast Preliminary to Electric Welding

ENAMELED IRON WORKS

Blowing or Spraying Enamel

FIRE STATIONS

Blowing Fire Whistles

Pumping Water
Cleaning with the Air Jet
Cleaning Boiler Flues

FOUNDRIES

Operating Sand Rammers and Molding Machines
Pneumatic Hammers and Drills
Pneumatic Sand Sifter
Air Hoists, Lifts and Motors
Cleaning Castings by the Sand Blast
Blowing Out Cores
Drilling Salamanders
Aerating Metal in Bessemer Process

GAS WORKS
Starting Gas Engines
Compressing Carbonic Acid
Acetylene, Oxygen and other Gases (Compressors)
Riveting and Calking Tanks
Calking and Testing Pipe Lines
Vaporizing Oil for Oil Engines
High Pressure Gas Transmission (Compressors)

GLASS WORKS
Pumping Glass Sand by the Return Air System
Blowing Glass
Operating Molds and Presses
Supplying Oil Burners
Operating Sand Blasts
Etching Glass

GRAIN ELEVATORS
Drying Grain

HAT FACTORIES
Operating Presses
Cleaning the Machines with the Air Jet

ICE AND REFRIGERATING PLANTS
Pumping and Aerating Water
Air Hoists for Ice Tanks
Air Hoist for Loading Cars
Scaling Condenser Coils
Cleaning Boiler Flues

IRRIGATION
Pumping Water by the Air Lift or Displacement Systems

JUTE MILLS
Pumping Water
Operating Air Hoists, Lifts and Motors

KNITTING MILLS
Pumping Water
Cleaning with the Air Jet
Agitating, Elevating and Transferring Solutions
Automatic Sprinkler System
Air Hoists, Lifts and Motors

MACHINE SHOPS
Pumping Water
Operating Pneumatic Hammers and Drills
Supplying Power Hammers, Air Hoists, Lifts and Jacks
Air Motors for Grinding, Buffing, etc.
Sand Blasts for Cleaning Metals
Cleaning Machines with the Air Jet
Belt Shifters for Heavy Machine Tools
Punches and Presses
Pipe Bending Machines

MINING
Running Rock Drills and Hammer Drills
Return-Air System for Station and Sump Pumping
Unwatering by the Air Lift System
Air Hoists and Motors
Pneumatic Tools for the Repair Shop
Operating Air Brakes on Hoists
Unloading Cars
Running Direct-Acting Pumps
Drill Steel Sharpeners
Blowing Smith Fires
Oil Furnaces for Steel Sharpeners
Pile Drivers for Shaft Work

MUNICIPALITIES AND PUBLIC INSTITUTIONS
Pumping Water by the Air Lift or Return-Air System
Pumping Sewage
Testing and Calking Pipe Lines
Rock Drills in Road Construction
Blowing Fire Whistles
Filtration Plants

OFFICE BUILDINGS AND HOTELS
Sewage and Drainage Ejectors
Pneumatic Clocks
Operating Elevator Doors
Cleaning with the Air Jet
Dentist, Physician and Barber Service
Accumulator Systems

OIL REFINERIES
Displacing and Transferring Oil
Agitating and Transferring Acids
Pneumatic Hammers and Drills
Calking Tanks and Pipe Lines
Air Hoists and Lifts
Oil Wells
Pumping by the Air Lift or Displacement Systems

PACKING HOUSE PLANTS
Pumping Water
Stuffing Sausages
Operating Belly Pounders

Lard Refineries
Air Hoists, Lifts and Motors

PAINT FACTORIES
Dressing Burr Stones
Agitating and Transferring Mixtures
Air Hoists and Lifts

PRINTING SHOPS
Monotype Machines
Air Hoists and Lifts
Cleaning by the Air Jet

QUARRIES
Rock Drills, Hammer Drills and Plug Drills
Stone Channelers
Air Hoists, Lifts and Motors
Drainage Pumping by the Return Air System
Operating Direct-Acting Pumps
Splitting Stone in the Bed

RAILROADS
Starting Fires in Locomotives
Operating Pneumatic Hammers and Drills
Cleaning Flues
Calking Boilers and Tanks
Air Hoists, Lifts and Jacks
Air Brakes
Turn-Table Motors
Crossing Gates
Switch and Signal Systems
Sand Blast for Removing Paint, Cleaning Castings, etc
Pneumatic Sand Rammers
Paint and Whitewash Spraying
Cleaning Cars, Furnishings, etc.
Transferring Oil
Pumping Water
Track Sander
Valve Setting Machines

ROAD BUILDING
Running Rock Drills and Plug Drills

RUBBER FACTORIES
Testing Rubber
Inflating Rubber Tires
Testing Tires, etc.
Air Hoists, Lifts and Motors

SALT MINES
Rock Drills and Plug Drills

SALT WELLS
Pumping Brine

SAND PITS
Pumping Sand by the Return-Air System

SAW MILLS
Operating Edgers, Bumpers, Pickups, etc.
Blowing Sawdust and Shavings from Machines
Pneumatic Haulage
Pumping Water
Automatic Sprinkler System

SEWAGE AND TRENCH WORK
Rock Drills and Hammer Drills
Sheet Pile Drivers
Calking and Testing Pipe Lines

SHIPYARDS AND BOAT BUILDERS
Pneumatic Hammers and Drills
Air Hoists, Lifts and Motors
Cleaning Plates and Castings by the Sand Blast
Pneumatic Painting
Starting Gasoline Engines
Riveting and Calking

SMELTERS AND ORE MILLS
Converter Tamping Machines
Air Hoists and Lifts
Blowing Converters
Agitating Cyanide Solution
Cleaning Cyanide Tanks by the Sand Blast
Calking Tanks
Handling Solutions

STEAM POWER PLANTS
Cleaning Boiler Flues
Calking Boilers
Pumping Water
Air Hoists and Lifts

STORE SERVICE
Pneumatic Cash Carriers
Cleaning by the Air Jet

STRUCTURAL WORKS
Pneumatic Tools for Drilling, Reaming, Chipping, Scaling, Riveting and Calking
Air Hoists
Pneumatic Punches
Rock Drills and Plug Drills
Drainage Pumps

SUGAR REFINERIES
Agitating and Transferring Syrups

Pumping Water
Air Hoists and Lifts
Pressure Filters

TANNERIES
Pumping Water
Handling Tan Liquid
Air Hoists and Lifts

THEATRES AND HALLS
Cleaning with the Air Jet
Displacing Water
Handling Scenery with the Air Hoist and Lift

U. S. GOVERNMENT
Pneumatic Tube Systems
Torpedo Chargings
Ammunition Hoists
Pumping Water
Sand Blasts

WAREHOUSES AND STORAGE
Air Hoists and Lifts
Air Motors
Air Cleaning

WATCH FACTORIES
Blowing Dust, Chips, etc., with the Air Jet
Pneumatic Tools
Air Hoists and Motors

WATER WORKS
Pumping Water
Cleaning Boiler Flues
Calking Boilers and Tanks
Calking and Testing Pipe Lines
Sheet Pile Drivers
Plug Drills and Rock Drills for Trenching

WOODWORKING MILLS
Bending Wood

WOOLEN MILLS
Pumping Water
Agitating and Handling Dyes and Solutions
Cleaning Looms, Doffers and Spindles
Air Motors, Hoists and Lifts
Pressure Accumulators
Starting Gas and GasolineEngines

Compressor Data Required for Full and Correct Reply

When writing for information or prices on compressors, you will insure a prompt, full and satisfactory reply by giving us as far as possible answers to the following questions:

Method of Drive — If Steam.

1. Maximum and minimum pressure at compressor throttle, or
2. Boiler pressure, distance from boiler to compressor, and size of steam pipe?
3. To operate condensing or non-condensing?
4. Are we to furnish condenser?
5. Have you any preference as to type of engine or steam valve to be used? Refer to cut in catalogue or advise in detail.
6. Special features or remarks?

If Electric.

7. Voltage of motor?
8. Direct current or alternating?
9. If alternating, phases and cycles?
10. Gear, chain or belt drive?
11. Slow, moderate or high-speed motor?
12. Have you any preference as to type or make of motor?
13. Special features or remarks?

Volume of Air Required.

14. Maximum air, cubic feet per minute?
    Free or compressed air? State which.
15. Maximum gauge pressure at machine, or
16. Maximum gauge pressure at work?
17. Distance from compressor to work?
18. Character of work to be done?
19. Altitude above sea level where compressor will be located?
20. Special features or remarks?

General Conditions.

21. Floor space and head room, if limited?
22. Are we to install?
23. Character of work for which required, such as gas compression, air lift pumping, handling oil or acids, etc.?
24. Approximate hours per day compressor will operate?
25. Comments and remarks?

Cost of Compressing Air

Coal, Gasoline, and Electric Current Compared

In the contemplation of purchasing a relatively small air compressor unit, say 30 to 100 H. P. capacity, it is often desirable to consider various sources of power for operating it and the cost of fuel. Data given in the Practical Reference Tables have been compiled to show readily these comparisons wherein the air compressor is to be driven either by:

1. Steam Engine, direct connected or belt driven.
2. Gasoline Engine, direct connected or belt driven.
3. Electric motor, direct connected or belt driven.

It may be noted from the example given in each table that to compress and deliver 100 cu. ft. of free air at 90 lb. pressure,

1. The cost of coal will be 0.5232 cent
2. The cost of gasoline will be 1.08 cents
3. The cost of electric current will be 1.1 cents

when the cost of coal is taken at $6.00 per ton, gasoline at 22 cents per gallon, and electric current furnished by a steam operated commercial distributing plant at 4 cents per K. W. Hour.

In using these tables, it is, of course, necessary to assume certain figures suiting local conditions in order to make proper comparisons, after first determining the horsepower required to compress and deliver 100 cu. ft. of free air per minute and assuming the mechanical efficiency of all the machines is about the same.

In Table I, the steam consumption of the engine must be approximately determined. In the example, it is taken at 34 lbs. per horsepower per hour; if it were 17 lbs., the cost of coal would be one-half. The boiler evaporation in the table is given as delivering to the engine 7 lbs. of water for each pound of coal burned, and any percentage of variation from this performance should appear in the calculations. When the final figure of cost in the table is determined, it must be multiplied by the cost of coal in dollars per ton. A steam driven compressor requires boiler power, and the cost of this and the expense of operating it must be considered, when comparing with other sources of power, aside from the cost of coal which latter is covered by the table.

In Table II, the gasoline consumption per brake horsepower will not vary greatly, but the percentage of variation should be considered. If kerosene or other oils are burned

instead of gasoline, the price per gallon should be determined, also the consumption per brake horsepower.

In Table III, it is assumed that the consumer may wish to buy electric current from some power plant.

The figures in all the tables are given in multiples so that missing ones can readily be determined; for illustration, if 100 cu. ft. of free air requires 10 H. P., one half the horizontal column of 30 or one-third of 30 can be taken. If gasoline costs 20 cents, multiply by 2 the column given under 10 cents. If electric current costs 6 cents, multiply by 2 the column given under 6 cents.

These tables can be used to advantage in considering a plant of any size or efficiency.

Note—In the cost figures the element of time need not be considered in compressing 100 cu. ft. of air, whether it requires one-quarter minute, one-half, etc., to do the work.

TABLE I

Cost of Coal at One Dollar Per Ton to Compress and Deliver 100 Cubic Feet of Free Air

(Multiply figures in table by price of coal in dollars per ton. Ton equals 2000 pounds).

Weight of Steam—in Pounds—Required by the Engine—Per Horsepower per Hour.

24	26	28	30	32	34	36	38	40	42

Weight of Coal—in Pounds—Per Horsepower per Hour—When Boiler is Evaporating 7 pounds of Water per Pound of Coal.

3.43	3.714	4.	4.286	4.571	4.757	5.143	5.428	5.714	6

Brake H. P. to deliver 100 cu. ft. free air per minute	Cost of Coal per Horsepower at One Dollar per Ton—Fractions of Cent.									
	.1715	.1857	.2	.2143	.2285	.2378	.2571	.2714	.2857	.3
	Cost of Coal per 100 Cubic Feet Free Air—Fractions of Cent.									
16	.0457	.0495	.0533	.0572	.0610	.0634	.0686	.0724	.0762	.08
18	.0514	.0557	.06	.0643	.0686	.0714	.0772	.0814	.0857	.09
20	.057	.0619	.0667	.0714	.0762	.0793	.0857	.0904	.0952	.10
22	.0627	.0681	.0733	.0785	.0838	.0872	.0943	.0995	.1047	.11
24	.0684	.0743	.08	.0857	.0914	.0951	.1029	.1085	.1142	.12
26	.0741	.0805	.0866	.0929	.099	.103	.1114	.1175	.1238	.13
28	.0798	.0867	.0933	.10	.1066	.1109	.12	.1266	.1334	.14
30	.0855	.0929	.10	.1071	.1142	.1189	.1285	.1357	.1429	.15

Example—If it requires 22 brake horsepower to deliver 100 cubic feet of free air per minute at 90 pounds pressure, and the compressor is driven by an ordinary steam engine requiring 34 pounds of steam per horsepower hour, and an ordinary boiler evaporating 7 pounds of water per pound of coal, and with coal at $6 a ton—the coal cost per 100 cubic feet of free air is .0872 x 6=.5232 cent.

TABLE II

Cost of Gasoline in Cents to Compress and Deliver 100 Cubic Feet of Free Air

Gasoline Consumption One Pint Per Brake Horsepower Per Hour

Brake H. P. to deliver 100 Cu. Ft. Free Air per Minute	Price of Gasoline per Gallon Cents							
	8	9	10	11	12	13	14	15
16	.266	.300	.333	.366	.400	.433	.465	.500
18	.300	.337	.375	.413	.450	.487	.525	.562
20	.333	.375	.416	.458	.500	.542	.584	.626
22	.366	.412	.458	.504	.550	.595	.642	.687
24	.400	.450	.500	.550	.600	.650	.700	.750
26	.433	.488	.542	.596	.650	.705	.759	.813
28	.466	.525	.583	.642	.700	.758	.816	.875
30	.500	.563	.625	.688	.750	.813	.875	.938

Example—If it requires 22 Brake Horsepower to deliver 100 cubic feet of free air per minute at 90 pounds pressure, and gasoline at 22 cents per gallon, the gasoline cost per 100 cubic feet of free air is .504x2=1.08 cents.

TABLE III

Cost of Electric Current in Cents to Compress and Deliver 100 Cubic Feet of Free Air

Brake H.P. to deliver 100 Cu. Ft. Free Air per Minute	Price of Electric Current per Kilowatt-Hour—Cents								
	1.	1.5	2.	2.5	3.	3.5	4.	4.5	5.
	Per Horsepower Hour—Cents								
	.75	1.125	1.5	1.875	2.25	2.625	3.	3.375	3.75
16	.2	.3	.4	.5	.6	.7	.8	.9	1.0
18	.225	.337	.45	.562	.675	.787	.9	1.012	1.125
20	.25	.375	.5	.625	.75	.875	1.0	1.125	1.25
22	.275	.412	.55	.687	.825	.962	1.1	1.237	1.375
24	.3	.45	.6	.75	.9	1.05	1.2	1.35	1.5
26	.325	.487	.65	.812	.975	1.137	1.3	1.46	1.625
28	.35	.525	.7	.875	1.05	1.225	1.4	1.575	1.75
30	.375	.562	.75	.937	1.125	1.312	1.5	1.687	1.875

Example—If it requires 22 Brake Horsepower to deliver 100 cubic feet of free air per minute at 90 pounds pressure, and electric current at 4 cents per kilowatt-hour, the electric current cost per 100 cubic feet of free air is 1.1 cents

H. V. CONRAD.

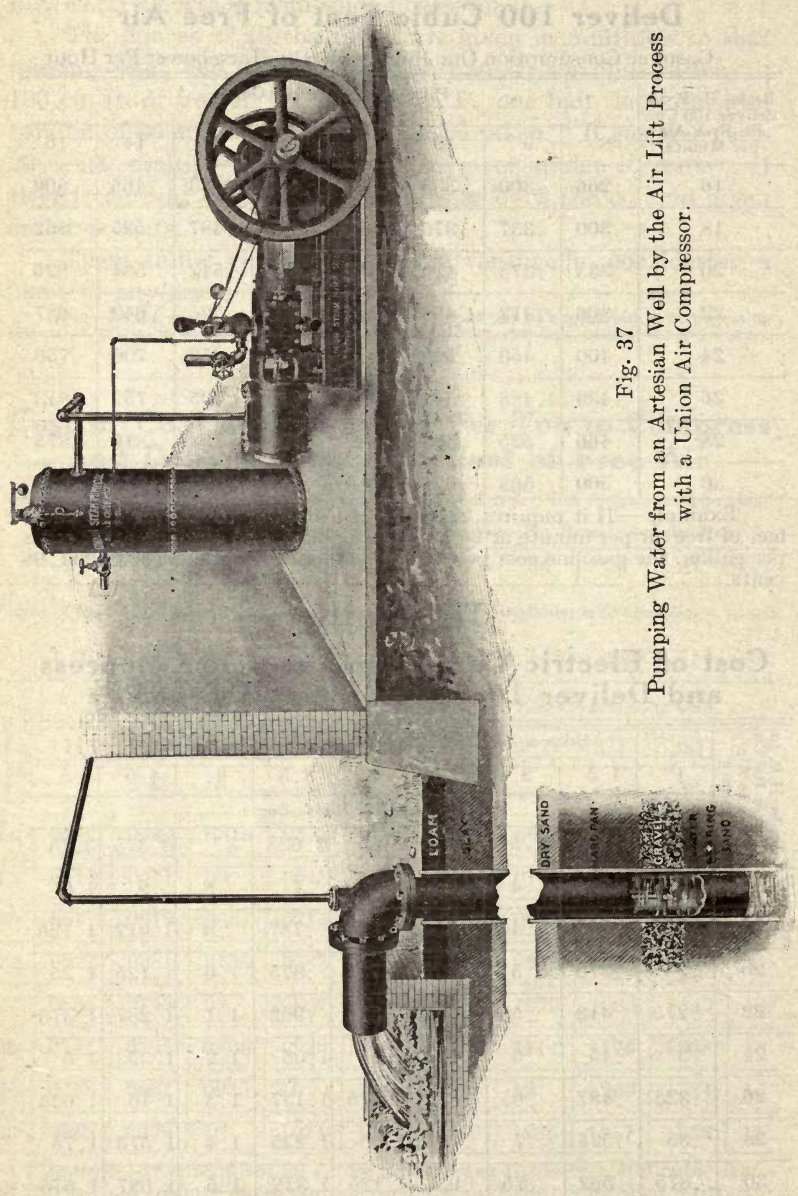


Fig. 37
Pumping Water from an Artesian Well by the Air Lift Process with a Union Air Compressor.

Pumping with Compressed Air

The air lift pumping system is a method of lifting water or other fluid by means of compressed air. This method of pumping is generally employed where conditions are such that it is impractical to use a mechanical pump.

Advantages

The principal advantages of the air lift system of pumping are the following:

Quantity: There is no question but that more water can be secured from a deep well with the air lift, than by any other method of pumping, providing the conditions are suitable for its use.

Quality: The aeration of the water produced by the air lift system of pumping is acknowledged to be one of the principal methods for purifying water in filtration plants.

If free sulphur gas is encountered in the well, it is almost completely removed by the action of the aeration in the air lift.

Temperature: With the air lift system of pumping, the expansion of the air lowers the temperature of the water.

Simplicity: The air lift is the simplest method of pumping, and it requires the least attention and repairs. With the air lift all the machinery is in the power house, and all trouble such as pulling sucker rods, working barrels, etc. is eliminated.

Terms Used in Air Lift Work

In discussing the air lift and air lift propositions, certain general terms are used that must be understood. By referring to figure 38, page 58, these terms are explained thusly:

Static Head: Normal water level, when not pumping, measured from the surface or ground level.

Drop: Point to which the water level drops below the static head, while being pumped.

Pumping Head: Level of water when pumping, with reference to the ground level. Pumping head equals static head plus the drop.

Elevation: Point above ground level to which the water is raised.

Total Lift: The distance water is raised, from level, when pumping, to point of discharge. Total Lift equals elevation plus static head plus drop.

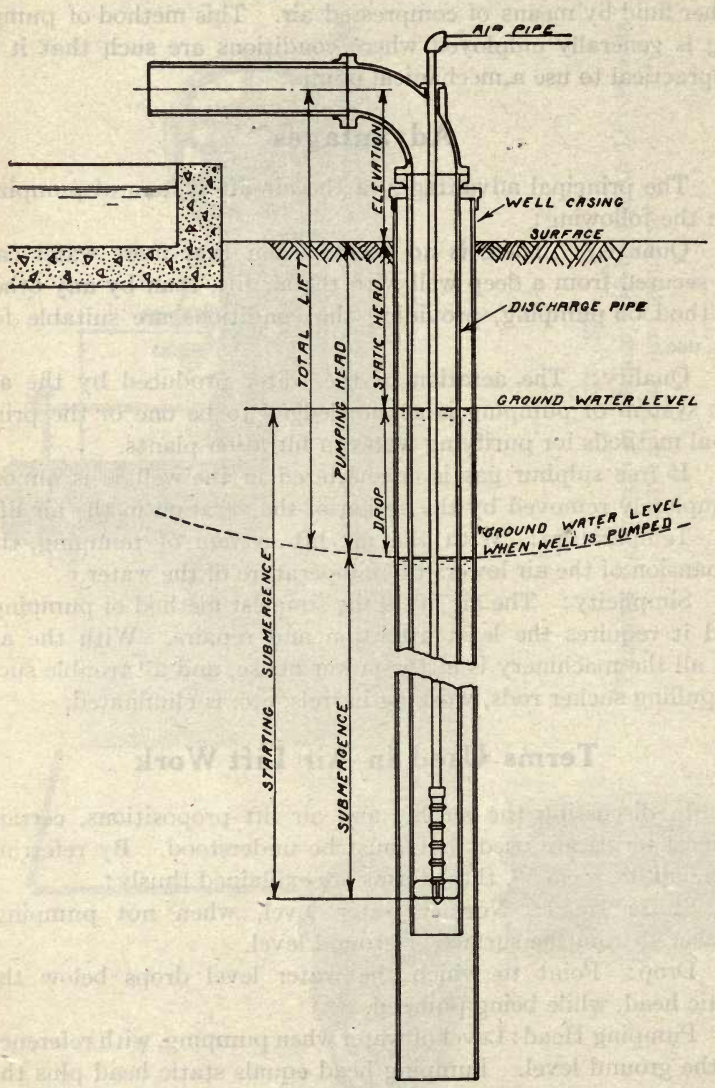


Fig. 38.
Diagram of an Air Lift.

Submergence: The distance below the pumping head at which the air picks up the water.

Starting Submergence: The distance below the static head at which the air picks up the water. Starting submergence equals drop plus submergence.

Ratio of Submergence

If you have a given ascertained percentage of submergence, the actual submergence would be ascertained by multiplying the lift by the percentage of submergence, and dividing the product by one hundred minus the percentage of submergence, as expressed in the following equation:

$$\text{Submergence} = \frac{\text{lift} \times \text{percentage of submergence.}}{100 - \text{percentage of submergence.}} \qquad (19)$$

The percentage or the ratio of submergence is expressed as follows:

$$\text{Ratio} = \frac{\text{submergence.}}{\text{lift} + \text{submergence.}} \qquad (20)$$

The necessary percentage of submergence varies with the lift; low lifts require proportionately more submergence than high lifts.

The following gives an idea of the proportion of submergence to lift, for good results:

For lifts up to	50 feet	66% submergence.
" " "	50 to 100 ft.	60% submergence
" " "	100 to 200 ft.	55% submergence
" " "	200 to 500 ft.	50% submergence.
" " "	300 to 400 ft.	45% submergence.
" " "	400 to 500 ft.	40% submergence.

The Air Lift Installation

To secure the best results, compressed air should be introduced into the well in a finely divided state, so that the bubbles are small and equally distributed throughout the water. It is evident that if the air pipe merely discharges the air into the water through a full opening in the pipe, the result will be large bubbles instead of the equally divided condition which is desired. It is therefore advisable to either cap the end of the air

pipe and drill small holes in the side near the end, or use an air nozzle as shown in figure 38, so that the air will be admitted to the water in small jets which will produce the small bubbles necessary for efficient results.

Inasmuch as the action of the air lift depends upon forming an emulsion of air and water, which is lighter than water, it is evident that a perfect condition would be one in which the bubbles, when introduced in the bottom of the well, would maintain the same size in their passage to the discharge. It will be readily seen, however, that inasmuch as the pressure is relieved from the air bubbles as they rise toward the surface, the bubbles become larger and larger; the proportion of the air to the water increases exactly in proportion to the expansion, and this decreases the efficiency of the lift.

This expansion of air, because it requires an increased volume per minute to pass through the pipe, has the effect of throttling the mixture of air and water, which is another source of loss in efficiency. The ideal lift would have its discharge line so proportioned that its area would be constantly increased. The proportion to this increase in volume thus keeps the velocity constant. In deep wells, it is advisable to approximate this condition by installing the discharge line in sections of different diameters.

Air and Water Pipes

The following table gives the sizes of water and air pipes required for the central pipe system of pumping as illustrated on page 58. The size of well casing and the pumping capacity for which these pipes are adapted is also given. The drop pipe should extend below the air nozzle not less than 5 feet, and from 15 to 20 feet, if possible, depending upon the depth of the well.

Smallest Well Casing	Water Pipe	Central Air Pipe	Capacity, Gallons Per Minute
2½	1½	½	10–20
3	2	¾	20–40
3½	2½	1	40–60
4½	3	1¼	60–80
5	3½	1¼	80–100
6	4	1½	100–150
7	5	2	150–250
8	6	2	275–375
10	8	2½	500–650
12	10	2½	775–1000

Calculation of the Air Lift.

We are now prepared to consider a concrete problem in air lift pumping. Given a well with a 7 inch casing, 300 feet deep, the water in which stands 75 feet below the ground level, but which falls 10 feet when being pumped at the rate of 200 gallons per minute, and it is required to raise the water 15 feet above the ground.

Then from the above·

Static Head equals.....75 feet

Drop equals.....10 feet.

Pumping head equals .....85 feet.

Elevation equals.....15 feet.

Total lift equals .....100 feet.

By referring to the table on page 59, the percentage of submergence for 100 feet lift is 60%. Substituting this value in equation 19, on page 59,

$$\frac{100 \times 60}{100 - 60} = \frac{6000}{40} = 150 \text{ feet submergence.}$$

With 150 feet submergence, and with the pumping head given of 85 feet, the distance from the surface of the ground to the bottom of the air pipe or nozzle will be,

$$150 + 85 = 235 \text{ feet.}$$

As stated on page 63, the drop pipe should (if possible), extend down 15 to 20 feet below the air pipe, or in the example given, 255 feet from the surface of the ground.

Referring to table on page 63, the compressor displacement required to elevate 200 gallons per minute against a total lift of 100 feet with 60% submergence is:

$200 \times .585 = 117.0$ cubic feet of free air per minute.

The indicated or brake horse power required to operate the compressor having 117 cubic feet displacement may be calculated by referring to the table for single stage compression, on page 38. The horse power per cubic foot displacement at 67½ pounds working pressure (interpolating between 65 pounds and 70 pounds) is .147 and

$$.147 \times 117 = 17.19 \text{ Ind. or B. H. P.}$$

required to operate the compressor.

The next calculation is to find the starting and working pressures. The pumping head is 10 feet greater than the static head, which indicates a starting submergence of 150 + 10 = 160 feet.

Starting pressure equals:

Starting submergence × .45

Working pressure equals:

Submergence × .45

Substituting the values of the above example,

The starting pressure equals:

160 × .45 = 72 pounds.

Working pressure equals

150 × .45 = 67½ pounds.

In the table on page 63, the working pressure is given, and the starting pressure may be calculated as above.

Then, referring to the table on page 60, for the conditions given in the example, it will require a 5-inch drop pipe, and a 2-inch air pipe.

In calculating an air lift, the pumping head in a well can seldom be known in advance of a test. It is customary to assume certain conditions of lift and submergence based on experience, and pipe the well accordingly. After the pipe is installed, the submergence is altered to suit, by raising or lowering the pipe in the well until the best results are obtained.

Approximate Cubic Feet of Free Air and Working Pressure Required to Raise One Gallon of Water by Air Lift*

$$\text{FORMULA} = \text{Log.}\frac{\frac{L}{H+34}}{34} \times 234$$

H = Submergence in Feet

L = Lift in Feet

RATIO OF SUBMERGENCE TO LIFT

Lift in Feet	25% ⅓ to 1		33% ½ to 1		43% ¾ to 1		50% 1 to 1		55% 1¼ to 1		60% 1½ to 1		66% 2 to 1		70% 2½ to1		75% 3 to 1	
	Free air Cubic Feet	Working Press	Free air Cubic Feet	Working Press	Free air Cubic Feet	Working Press	Free air Cubic Feet	Working Press	Free air Cubic Feet	Working Press	Free air Cubic Feet	Working Press	Free air Cubic Feet	Working Press.	Free air Cubic Feet	Working Press	Free air Cubic Feet	Working Press
20	.....	.....	.....	.....	.....	.....	.428	9	.365	11	.310	13½	.252	18	.217	22½	.195	27
30	.....	.....	.....	.....	.....	.....	.470	13½	.400	17	.350	20	.290	27	.255	34	.230	40½
40	.....	.....	.....	.....	.....	.....	.508	18	.435	22½	.387	27	.325	36	.287	45	.263	54
50	.....	.....	.....	.....	.....	.....	.546	22½	.470	28	.422	34	.360	45	.320	56	.285	67½
60	.....	.....	.....	.....	.....	.....	.582	27	.510	34	.457	40½	.392	54	.350	67½	.323	81
80	.....	.....	.....	.....	.....	.....	.653	36	.575	45	.522	54	.455	72	.410	90	.380	108
100	.....	.....	.....	.....	.850	34	.720	45	.640	56	.585	67½	.512	90	.465	112½	.433	135
120	.....	.....	.....	.....	.915	40½	.785	54	.703	67½	.642	81	.567	108	.520	135	.483	162
140	.....	.....	.....	.....	.982	47	.847	63	.763	79	.697	94½	.622	126	.572	157½	.533	189
160	.....	.....	.....	.....	1.047	54	.907	72	.820	90	.755	108	.675	144	.624	180	.583	216
180	.....	.....	.....	.....	1.107	61	.965	81	.875	101	.810	121½	.725	162	.672	202	.630	243
200	1.8	30	1.436	45	1.167	67½	1.022	90	.930	112½	.862	135	.775	180	.720	225	.675	270
250	1.96	38	1.592	56	1.312	84	1.156	112½	1.069	141	.988	168	.895	225	.830	282	.....	.....
300	2.12	45	1.750	67½	1.507	100	1.292	135	1.206	169	1.110	200	1.010	270	.942	338	.....	.....
350	2.28	53	1.897	79	1.635	118	1.449	157½	1.317	197	1.232	236	1.110	315	1.050	394	.....	.....
400	2.45	60	2.045	90	1.725	135	1.542	180	1.429	225	1.345	270	1.232	360	1.155	450	.....	.....
450	2.60	68	2.182	100	1.850	152	1.669	205	1.542	253	1.457	304	1.337	410	1.280	506	.....	.....
500	2.74	75	2.328	112½	1.952	169	1.790	225	1.657	281	1.570	338	1.442	450	1.356	562	.....	.....
550	2.88	83	2.455	128	2.105	185½	1.907	247½	1.772	309	1.678	371	1.564	495	.....	.....	.....	.....
600	3.02	90	2.564	135	2.225	205	2.018	270	1.884	337½	1.784	410	1.642	540	.....	.....	.....	.....
650	3.16	98	2.730	146	2.345	219	2.175	292½	2.012	365½	1.908	438	1.742	585	.....	.....	.....	.....
700	3.31	105	2.845	157½	2.460	236	2.258	315	2.100	394	2.000	472	1.842	630	.....	.....	.....	.....
750	3.45	113	2.970	169	2.576	256	2.353	337½	2.225	424	2.080	512	1.935	675	.....	.....	.....	.....
800	3.60	120	3.095	180	2.690	270	2.465	360	2.320	450	2.190	540	2.063	720	.....	.....	.....	.....
850	3.72	128	3.215	191	2.800	287	2.576	382½	2.410	478	2.29	574	2.125	765	.....	.....	.....	.....
900	3.85	135	3.338	202½	2.915	304	2.675	405	2.50	506	2.325	608	2.220	810	.....	.....	.....	.....
950	3.98	142	3.455	214	3.0	322	2.780	427½	2.610	535	2.485	644	2.310	854	.....	.....	.....	.....
1000	4.11	150	3.575	225	3.465	337½	2.885	450	2.710	562½	2.580	675	2.407	900	.....	.....	.....	.....

*Cubic feet of Free Air given is the compressor displacement.

Information Required for Correct Recommendation for Air-Lift Pumping

Air Lift Pumping.

Number of wells and relative locations, distance apart and distance from proposed location of compressor. Also the following for each:

1. Entire depth of well.
2. Inside diameter.
3. If boring is reduced, state at what depth and to what diameter.
4. If cased, state inside diameter of casing.
5. Depth from surface to water level when not pumping.
6. Capacity of well at present when pumped in gallons per minute.
7. Depth from surface to water level when pumped at this capacity.
8. Style, kind, size, and capacity of pump at present used.
9. Elevation above ground surface to which water is to be raised.
10. Horizontal distance from well to tank.
11. Is compressor to be operated by steam direct, electric motor or belt power?
12. If by steam direct, state steam pressure carried.
13. At what depth below the surface is the source of water located?
14. Number of gallons per minute required.

Displacement in Cubic Feet Per Minute, Piston Working in Single Cylinder at Various Piston Speeds

Diam. of Cylinder Inches	PISTON SPEED, FEET PER MINUTE																	
	100	150	200	225	250	275	300	325	350	375	400	425	450	475	500	550	600	700
3	4.9	7.4	9.8	11.1	12.3	13.5	14.7	16	17.2	18.5	19.6	21	22.2	23.4	24.6	27	29.4	34.4
4	8.7	13.1	17.4	19.7	21.8	24	26.2	28.5	30.7	32.9	34.8	37.2	39.4	41.5	43.6	48	52.4	61.4
5	13.6	20.5	27.2	30.8	34.2	37.5	41	44.5	48	51.4	54.4	58	61.6	65	68.4	75	82	96
6	19.6	29.5	39.2	44.4	49	54	59	64	69	74	78.4	84	88.8	93	98	108	118	138
7	26.7	40	53.4	60.5	67	74	80	87	94	100.6	106.8	114	121	127	134	148	160	188
8	35	52.5	70	79	87.5	96	105	114	123	132	140	149	158	166	175	192	210	246
9	44	66.5	88	100	110	122	133	144	155	166	176	188	200	210	220	244	266	310
10	54.5	82	109	123	136	150	164	178	192	206	218	233	246	260	272	300	328	384
11	66	99	132	149	165	182	198	215	232	250	264	282	298	315	330	364	396	464
12	78	118	156	177	196	216	236	256	276	296	312	335	354	374	392	432	472	552
13	92	138	184	208	230	255	276	300	325	345	368	392	416	440	460	510	552	650
14	106	160	212	240	266	295	320	348	375	400	424	455	480	510	532	590	640	750
15	122	184	244	276	305	337	368	400	432	460	488	521	552	580	610	674	736	864
16	139	210	278	315	350	385	420	455	490	525	556	595	630	660	700	790	840	980
17	157	236	314	355	395	435	472	512	550	590	628	672	710	750	790	870	944	1100
18	176	265	352	400	440	485	530	575	620	660	704	755	800	840	880	970	1060	1240
19	196	295	392	445	490	540	590	640	690	740	784	840	890	930	980	1080	1180	1380
20	218	328	436	490	545	600	656	710	765	820	872	930	980	1040	1090	1200	1312	1530
22	265	395	530	595	660	725	790	860	930	990	1060	1120	1190	1260	1320	1450	1580	1860
24	314	470	628	710	790	865	940	1020	1100	1180	1256	1340	1420	1500	1580	1730	1880	2200
26	369	550	738	830	920	1020	1100	1200	1300	1390	1476	1570	1660	1750	1840	2040	2200	2600
28	427	640	854	960	1070	1180	1280	1390	1500	1600	1780	1820	1920	2030	2140	2360	2560	3000
30	490	740	980	1110	1230	1350	1480	1600	1720	1840	1960	2100	2220	2340	2460	2700	2960	3440
32	560	835	1120	1260	1400	1540	1670	1820	1960	2100	2240	2380	2520	2660	2800	3080	3340	3920
34	630	945	1260	1420	1570	1740	1890	2050	2220	2360	2520	2690	2840	3000	3140	3480	3780	4440
36	710	1060	1420	1600	1770	1950	2120	2300	2500	2650	2840	3000	3200	3360	3540	3900	4240	5000
38	780	1180	1560	1780	1960	2170	2360	2560	2750	2950	3120	3350	3560	3750	3920	4340	4720	5500
40	870	1310	1740	1970	2180	2400	2620	2850	3060	3280	3480	3720	3940	4150	4360	4800	5240	6120

Horse-Power, Efficiency and Terminal Temperature in Air Compression to Various Pressures

Single and Two Stage Compression

Gauge Pressure	Atmospheres	Isothermal Compression	Single Stage Adiabatic Compression			Two Stage Compression		
		H. P. required to Compress One Cubic Foot per Minute	H. P. required to Compress One Cubic Foot Free Air per Minute	Efficiency as compared to Isothermal	Final Temperature Degrees Fahr. Perfect Intercooling Adiabatic Compression	H. P. required to Compress One Cubic Foot Free Air per Minute	Efficiency as compared to Isothermal	Final Temperature Degrees Fahr. Perfect Intercooling Adiabatic Compression
5	1.34	.0188	.0197	.96	106			
10	1.68	.0333	.0362	.93	145			
15	2.02	.0481	.0505	.90	178			
20	2.36	.0551	.0630	.88	207			
25	2.70	.0638	.075	.85	234			
30	3.04	.0713	.085	.84	252			
40	3.72	.0843	.104	.81	302			
50	4.40	.0948	.120	.79	339	.109	.87	188
60	5.08	.1037	.134	.77	375	.121	.86	203
70	5.76	.1120	.148	.75	405	.131	.85	214
80	6.44	.1196	.160	.74	432	.141	.85	224
90	7.12	.1260	.171	.74	459	.150	.86	234
100	7.80	.1320	.182	.73	485	.158	.83	243
110	8.48	.1371	.192	.72	500	.165	.83	250
120	9.16	.1422	.202	.71	529	.172	.83	257
130	9.84	.1467	.210	.70	560	.179	.82	265
140	10.52	.1510	.218	.69	570	.186	.82	272
150	11.20	.1547	.226	.69	589	.193	.81	279
160	11.88	.1583	.234	.68	607	.198	.81	285
180	13.24	.1656	.249	.67	640	.208	.80	297
200	14.60	.1720	.263	.65	672	.217	.79	309
225	16.3	.1790	.278	.64	715	.227	.79	320
250	18.	.1860	.292	.64	749	.237	.78	331
275	19.7	.1920	.306	.63	780	.247	.78	342
300	21.4	.1970	.317	.62	815	.256	.77	352
350	24.8	.2060	.342	.60	867	.272	.76	370
400	28.2	.2140	.364	.59	915	.283	.76	380
450	31.6	.2230	.381	.58	960	.295	.75	397
500	35.	.2290	.398	.57	1000	.307	.75	415
550	38.4	.2340	.416	.56	1040	.321	.73	430
600	41.8	.240	.432	.55	1077	.332	.72	442
650	45.2	.245	.447	.55	1113	.345	.71	451
700	48.6	.249	.461	.54	1136	.355	.70	458
750	52.	.252	.475	.53	1178	.363	.69	462
800	55.4	.258	.488	.52	1208	.373	.69	468
850	58.8	.262	.500	.52	1237	.381	.69	480
900	62.2	.265	.512	.52	1265	388	.68	490
950	65.6	.268	.523	.51	1292	.395	.68	495
1000	69.	.272	.534	.51	1318	.403	.67	498
1100	75.8	.278	.555	.50	1367	.416	.67	507
1200	82.6	.283	.575	.49	1415	.429	.66	525
1300	89.4	.289	.594	.49	1457	.441	.66	534
1400	96.2	.293	.611	.48	1498	.452	.65	550
1500	103.	.297	.627	.48	1537	.462	65	563
1600	109.8	.301	.643	.47	1575	.472	.64	568
1700	116.6	.305	.659	.47	1610	.482	.63	589
1800	123.4	.309	.673	.46	1645	.491	.63	606
1900	130.2	.313	.687	46	1678	.500	.63	628
2000	139.	.317	.701	.45	1709	.509	.62	639
2250	154.	.324	.733	.44	1784	528	.62	645
2500	171.	.331	.763	.43	1852	.547	.61	654
3000	205.	.342	.816	42	1975	.579	.59	670

Horse-Power, Efficiency and Terminal Temperature in Air Compression to Various Pressures

Three and Four Stage Compression

Gauge Pressure	Atmospheres	Isothermal Compression	Three Stage Compression			Four Stage Compression		
		H. P. required to Compress One Cubic Foot per Minute	H. P. required to Compress One Cubic Foot Free Air per Minute	Efficiency as compared to Isothermal	Final Temperature Degrees Fahr. Perfect Intercooling Adiabatic Compression	H. P. required to Compress One Cubic Foot Free Air per Minute	Efficiency as compared to Isothermal	Final Temperature Degrees Fahr. Perfect Intercooling Adiabatic Compression
5	1.34	.0188						
10	1.68	.0333						
15	2.02	.0481						
20	2.36	.0551						
25	2.70	.0638						
30	3.04	.0713						
40	3.72	.0843						
50	4.40	.0948						
60	5.08	.1037						
70	5.76	.1120						
80	6.44	.1196						
90	7.12	.1260						
100	7.80	.1320						
110	8.48	.1371						
120	9.16	.1422						
130	9.84	.1467						
140	10.52	.1510						
150	11.20	.1547	.182	.85	200			
160	11.88	.1583	.187	.85	204			
180	13.24	.1656	.197	.84	211			
200	14.60	.1720	.206	.83	218			
225	16.3	.1790	.215	.83	224			
250	18.	.1860	.224	.83	230			
275	19.7	.1920	.233	.82	236			
300	21.4	.1970	.241	.82	241			
350	24.8	.2060	.252	.82	250			
400	28.2	.2140	.262	.82	258			
450	31.6	.2230	.272	.82	266			
500	35.	.2290	.282	.81	275	26.	.88	215
550	38.4	.2340	.292	.80	283	26.9	.87	220
600	41.8	.240	.300	.80	290	27.8	.86	225
650	45.2	.245	.310	.79	295	28.4	.86	228
700	48.6	.249	.320	.78	300	29.	.86	234
750	52.	.252	.327	.78	305	29.6	.85	236
800	55.4	.258	.334	.78	309	30.2	.85	240
850	58.8	.262	.341	.77	314	30.7	.85	244
900	62.2	.265	.347	.76	319	31.2	.85	247
950	65.6	.268	.354	.76	322	31.6	.85	250
1000	69.	.272	.360	.75	325	32.	.85	252
1100	75.8	.278	.370	.75	331	32.7	.85	254
1200	82.6	.283	.381	.74	338	33.4	.84	258
1300	89.4	.289	.390	.74	342	34.1	.84	265
1400	96.2	.293	.399	74	349	34.8	.84	270
1500	103.	.297	.406	73	353	35.5	.84	273
1600	109.8	.301	.415	.73	358	36.1	.83	276
1700	116.6	.305	.424	.72	364	36.7	.83	280
1800	123.4	.309	.431	.72	370	37 2	.83	284
1900	130.2	.313	.438	.72	374	37.7	.83	287
2000	139.	.317	.444	.71	378	38.1	.83	290
2250	154.	.324	.460	.70	385	39.3	.82	294
2500	171.	.313	.474	.70	398	40.5	.82	298
3000	205.	.342	.500	.69	414	42.	.81	308

Loss of Work Due to Heat in Compressing Air From Atmospheric Pressure to Various Gauge Pressures by Simple and Compound Compression

Air in Each Cylinder; Initial Temperature 60° F.

Gauge Pressure	One Stage		Two Stage		Three Stage		Four Stage	
	Percentage of Work Lost in Terms of							
	Isothermal Compression	Adiabatic Compression	Isothermal Compression	Adiabatic Compression	Isothermal Compression	Adiabatic Compression	Isothermal Compression	Adiabatic Compression
60	29.9	23.0	13.4	11.8	8.6	7.9	4.7	4.5
70	30.6	23.4	14.1	12.4	8.7	8.0	6.1	5.7
80	32.7	24.6	14.7	12.8	9.7	8.9	6.4	6.0
90	34.7	25.8	16.1	13.8	10.5	9.5	7.3	6.8
100	36.7	26.8	16.9	14.5	10.9	9.8	7.8	7.3
125	41.1	29.2	18.5	15.6	11.6	10.4	8.8	8.1
150	44.8	30.9	20.1	16.7	12.3	10.9	9.1	8.4
200	51.2	33.9	22.2	18.1	14.0	12.3	10.5	9.5
300	61.2	37.9	25.7	20.5	16.6	14.2	12.0	10.7
400	68.7	40.7	28.9	22.4	18.2	15.4	13.1	11.5
500	70.6	41.4	31.2	23.8	19.3	16.2	14.1	12.3
600	80.4	44.5	32.8	24.7	20.4	16.9	14.9	13.0
700	85.0	46.0	34.6	25.7	21.3	17.6	16.1	13.8
800	89.5	47.2	35.7	26.3	22.0	18.1	16.2	13.9
900	93.0	48.2	37.1	27.0	22.6	18.5	16.6	14.4
1000	96.1	49.0	27.9	27.5	23.2	18.8	16.9	14.5
1200	102.8	50.7	40.3	28.8	24.8	19.9	17.7	15.0
1400	108.6	52.0	41.5	29.3	25.9	20.5	18.6	15.7
1600	113.4	53.1	43.5	30.3	26.5	20.9	19.2	16.1
1800	117.5	54.0	44.8	31.0	27.3	21.2	19.6	16.4
2000	122.0	55.0	45.8	31.4	27.5	21.5	19.9	16.5

E. F. SCHAEFER.

Table of Volumes, Mean Pressures, Temperatures, Etc.

IN THE OPERATION OF

Air Compression from One Atmosphere and 60 Deg. F.

Gauge Pressure	Absolute Pressure	Pressure in Atmospheres	Volume with Air at Constant Temperature	Volume with Air not Cooled	Mean Pressure per Stroke. Air at Constant Temperature	Mean Pressure per Stroke. Air not cooled	Final Temperatures. Air not cooled	Gauge Pressure
0	14.7	1.	1.	1.	0.	0.	60	0
5	19.7	1.34	.7462	.81	4.3	4.5	106	5
10	24.7	1.68	.5952	.69	7.62	8.27	145	10
15	29.7	2.02	.495	.606	10.33	11.51	178	15
20	34.7	2.36	.4237	.543	12.62	14.4	207	20
25	39.8	2.7	.3703	.494	14.59	17.01	234	25
30	44.7	3.04	.3289	.4638	16.34	19.4	255	30
35	49.7	3.381	.2957	.42	16.92	21.6	281	35
40	54.7	3.721	.2687	.393	19.32	23.66	302	40
45	59.7	4.061	.2462	.37	20.52	25.59	321	45
50	64.7	4.401	.2272	.35	21.79	27.39	339	50
55	69.7	4.741	.2109	.331	22.77	29.11	357	55
60	74.7	5.081	.1968	.3144	23.84	30.75	375	60
65	79.7	5.423	.1844	.301	24.77	31.69	389	65
70	84.7	5.762	.1735	.288	26.	33.73	405	70
75	89.7	6.102	.1639	.276	26.65	35.23	420	75
80	94.7	6.442	.1552	.267	27.33	36.6	432	80
85	99.7	6.782	.1474	.2566	28.05	37.94	447	85
90	104.7	7.122	.1404	.248	28.78	39.18	459	90
95	109.7	7.462	.134	.24	29.53	40.4	472	95
100	114.7	7.802	.1281	.232	30.07	41.6	485	100
105	119.7	8.142	.1228	.2254	30.81	42.78	496	105
110	124.7	8.483	.1178	.2189	31.39	43.91	507	110
115	129.7	8.823	.1133	.2129	31.98	44.98	508	115
120	134.7	9.163	.1091	.2073	32.54	46.04	529	120
125	139.7	9.503	.1052	.202	33.07	47.06	540	125
130	144.7	9.843	.1015	.1969	33.57	48.1	550	130
135	149.7	10.183	.0981	.1922	34.05	49.1	560	135
140	154.7	10.523	.098	.1878	34.57	50.02	570	140
145	159.7	10.846	.0921	.1837	35.09	51.	580	145
150	164.7	11.204	.0892	.1796	35.48	51.89	589	150

Efficiencies of Air Compression at Different Altitudes

Altitude	Barometric Pressure		Volumetric Efficiency Compressor Per Cent	Loss of Capacity Per Cent	Decreased Power Required Per Cent
	Inches Mercury	Pounds per Sq. Inch			
0	30.00	14.75	100	0	0.
1000	28.88	14.20	97	3	1.8
2000	27.80	13.67	93	7	3.5
3000	26.76	13.16	90	10	5.2
4000	25.76	12.67	87	13	6.9
5000	24.79	12.20	84	16	8.5
6000	23.86	11.73	81	19	10.1
7000	22.97	11.30	78	22	11.6
8000	22.11	10.87	76	24	13.1
9000	21.29	10.45	73	27	14.6
10000	20.49	10.07	70	30	16.1
11000	19.72	9.70	68	32	17.6
12000	18.98	9.34	65	35	19.1
13000	18.27	8.98	63	37	20.6
14000	17.59	8.65	60	40	22.1
15000	16.93	8.32	58	42	23.5

Density of Gases and Vapors

Compared with air at same temperature and pressure; also weight of a cubic foot at 62° F. under atmospheric pressure of 14.7 lbs. abs. or 29.92 inches mercury.

	Density, Air at same temp. and pres. being 1.0 (Regnault)	Specific Gravity or Density, Water at 62° being 1.0	Wt. of a Cu. Foot in Pounds	Cubic Feet at 62° in One Pound
Air (atmospheric)	1.00000	.001221 or $\frac{1}{819}$	.07610	13.14
Hydrogen gas	.06926	.0000846 or $\frac{1}{11820}$	.00527	189.70
Oxygen gas	1.10563	.001350 or $\frac{1}{741}$	.08414	11.88
Nytrogen gas	.97137	.001185 or $\frac{1}{844}$	.07383	13.54
Carbonic acid gas	1.52901	.001870 or $\frac{1}{535}$	.11636	8.59
Carbonic oxide gas	.9674	.00118 or $\frac{1}{847}$	.07364	13.60
Vapor of water	.6235.	.0007613 or $\frac{1}{1313}$	.04745	21.07
Vapor of alcohol	1.589	.00194 or $\frac{1}{515}$	.12092	8.27
Vapor of sulphuric ether	2.586	.00316 or $\frac{1}{316}$	.19680	5.08
Vapor of oil of turpentine	4.760	.00581 or $\frac{1}{172}$	.36224	2.76
Vapor of mercury	6.976	.00850 or $\frac{1}{118}$	.52987	1.88

Effect of Initial or Intake Temperature on Efficiency and Capacity of Air Compressors

Unit Capacity and Efficiency Assumed at 60° F.

Initial Temperature		Relative Capacities and Efficiencies	Initial Temperature		Relative Capacities and Efficiencies
Degrees Fahr.	Degrees Abs.		Degrees Fahr.	Degrees Abs.	
20	441	1.18	70	583	.980
10	451	1.155	80	541	.961
0	461	1.13	90	551	.914
10	471	1.104	100	561	.928
20	481	1.083	110	571	.912
30	491	1.061	120	581	.896
32	493	1.058	130	591	.880
40	501	1.040	140	601	.866
50	511	1.020	150	611	.852
60	521	1.000	160	621	.838

Multipliers to be Used for Transforming Volumes of Compressed Air at Various Pressures Into Corresponding Volumes of Free Air at Atmospheric Pressure of 14.7 Pounds

Pressure, Lbs.	Multiplier	Pressure, Lbs.	Multiplier	Pressure, Lbs.	Multiplier	Pressure, Lbs.	Multiplier	Pressure, Lbs.	Multiplier
1	1.068027	26	2.768602	51	4.469377	76	6.170052	101	7.870727
2	1.136054	27	2.836729	52	4.537404	77	6.238079	102	7.938754
3	1.204081	28	2.904756	53	4.605431	78	6.306106	103	8.006781
4	1.272108	29	2.927783	54	4.673458	79	6.374133	104	8.074808
5	1.340135	30	3.040810	55	4.741485	80	6.442160	105	8.142835
6	1.408162	31	3.108837	56	4.809512	81	6.510187	106	8.210862
7	1.476189	32	3.176864	57	4.877539	82	6.578214	107	8.278889
8	1.544216	33	3.244891	58	4.945566	83	6.646241	108	8.346916
9	1.612243	34	3.312918	59	5.013593	84	6.714268	109	8.414943
10	1.680270	35	3.380945	60	5.081620	85	6.782295	110	8.482970
11	1.748297	36	3.448972	61	5.149647	86	6.850322		
12	1.816324	37	3.516999	62	5.217674	87	6.918349		
13	1.884351	38	3.585026	63	5.285701	88	6.986376		
14	1.952378	39	3.653053	64	5.353728	89	7.054403		
15	2.020405	40	3.721080	65	5.421755	90	7.122430		
16	2.088432	41	3.789107	66	5.489782	91	7.190457		
17	2.156459	42	3.857134	67	5.557809	92	7.258484		
18	2.224486	43	3.925161	68	5.625836	93	7.326511		
19	2.292513	44	3.993188	69	5.693863	94	7.394538		
20	2.360540	45	4.061215	70	5.761890	95	7.462565		
21	2.428567	46	4.129242	71	5.829917	96	7.530592		
22	2.496594	47	4.197269	72	5.897944	97	7.598619		
23	2.564621	48	4.265296	73	5.965971	98	7.666646		
24	2.632648	49	4.333323	74	6.033998	99	7.734673		
25	2.700675	50	4.401350	75	6.102025	100	7.802700		

Formula $\frac{1}{14.7} \times$ Pressure $+ 14.7$ = Multiplier.

Atm. Press. 14.7 pounds = 30″ Barom. Press. Temp. 60° F.

Table for Single Compression at Altitudes

Altitude in Feet	Barometric Pressure		Per Cent of Efficiency of Air Considering its Volume 100 Per Cent at Sea Level		Per Cent Increase in Volume Necessary to give at Altitudes Efficiency Equivalent to Sea Level		Horse Power Required to Compress One Cubic Foot of Free Air at Different Altitudes (Single Compression)		Per Cent of Power Required to Compress Same Piston Displacement Compared with Sea Level		Per Cent Decrease in Power Required to Compress Same Piston Displacement Compared with Sea Level		Per Cent Increase in Power Required to Compress the Volume of Air at Altitude Equal to a Given Volume at Sea Level	
	Inches of Mercury	Pounds per Square Inch	Gauge		Gauge		Gauge		Gauge		Gauge		Gauge	
			80	100	80	100	80	100	80	100	80	100	80	100
0	30.00	14.75	100.0	100.0	0.0	0.0	.1585	.1800	100.0	100.0	0.0	0.0	0.0	0.0
1000	28.88	14.20	96.8	96.7	3.3	3.4	.1561	.1773	98.5	98.4	1.5	1.6	1.7	1.8
2000	27.80	13.67	93.8	93.6	6.6	6.8	.1536	.1744	96.9	96.8	3.1	3.2	3.3	3.4
3000	26.76	13.16	90.7	90.5	10.2	10.5	.1512	.1714	95.4	95.2	4.6	4.8	5.1	5.2
4000	25.76	12.67	87.8	87.5	13.9	14.2	.1486	.1685	93.7	93.6	6.3	6.4	6.8	6.9
5000	24.79	12.20	84.9	84.6	17.6	18.2	.1464	.1657	92.4	92.0	7.6	8.0	8.6	8.8
6000	23.86	11.73	82.1	81.7	21.8	22.4	.1438	.1629	90.7	90.5	9.3	9.5	10.6	10.7
7000	22.97	11.30	79.5	78.9	25.7	26.7	.1416	.1599	89.3	88.8	10.7	11.2	12.3	12.6
8000	22.11	10.87	76.9	76.3	30.0	31.0	.1391	.1571	87.7	87.3	12.3	12.7	14.1	14.3
9000	21.29	10.46	74.3	73.7	34.7	35.6	.1369	.1543	86.4	85.7	13.6	14.3	16.2	16.3
10000	20.49	10.07	71.9	71.2	39.0	40.0	.1348	.1515	85.0	84.1	15.0	15.9	18.2	18.3

Table for Two-Stage Compression at Altitudes

Altitude in Feet	Barometric Pressure		Percentage of Efficiency of Air at Altitudes Considering its Volume 100 Per Cent at Sea Level				Percentage Increase in Volume Necessary to give at Altitudes Efficiency Equivalent to Sea Level				Horsepower Required to Compress One Cubic Foot of Free Air to			
			GAUGE PRESSURE				GAUGE PRESSURE				GAUGE PRESSURE			
	Inches of Mercury	Pounds per Sq. Inch	60	80	100	125 150	60	80	100	125 150	60	80	100	125 150
														.1708
0	30.00	14.75	100.0	100.0	100.0	100.0	0.0	0.0	0.0	0.0	.1175	.1375	.1535	.1859
						96.6				3.5				.1675
1,000	28.88	14.20	96.9	96.8	96.7	96.5	3.2	3.3	3.4	3.6	.1157	.1345	.1508	.1820
						93.4				7.0				.1641
2,000	27.80	13.67	94.0	93.8	93.6	93.3	6.3	6.6	6.8	7.2	.1143	.1324	.1481	.1781
						90.2				11.0				.1607
3,000	26.76	13.16	91.2	90.7	90.5	90.0	9.6	10.2	10.5	11.1	.1116	.1298	.1449	.1742
						87.2				14.6				.1574
4,000	25.76	12.67	88.3	87.8	87.5	87.0	13.2	13.9	14.2	14.9	.1097	.1273	.1422	.1707
						84.2				18.7				.1541
5,000	24.79	12.20	85.6	84.9	84.6	84.0	16.8	17.6	18.2	19.0	.1079	.1251	.1391	.1670
						81.2				23.1				.1510
6,000	23.86	11.73	82.8	82.1	81.7	81.0	20.7	21.8	22.4	23.4	.1059	.1228	.1362	.1634
						78.5				27.4				.1478
7,000	22.97	11.30	80.3	79.5	78.9	78.2	24.5	25.7	26.7	27.8	.1040	.1203	.1336	.1598
						75.8				31.9				.1445
8,000	22.11	10.87	77.7	76.9	76.3	75.4	28.7	30.0	31.0	32.6	.1020	.1180	.1308	.1563
						73.1				36.8				.1415
9,000	21.29	10.46	75.2	74.3	73.7	72.8	32.9	34.7	35.6	37.3	.1003	.1154	.1281	.1529
						70.6				41.6				.1384
10,000	20.49	10.07	72.8	71.9	71.2	70.2	37.3	39.0	40.0	42.4	.0984	.1133	.1254	.1493

Compressed Air Table for Hoisting Engines

The following table is intended to give an approximate idea of the volume of free air required for operating hoisting engines, the air being delivered to the engines at 60 pounds gauge pressure. There are so many variable conditions in the operation of hoisting by the hoisting engines in common use, that accurate computations can only be offered when fixed data are given. In the table, the hoisting engine is assumed to actually run but one-half of the time for hoisting, while the compressor, of course, runs continuously. If the engine runs less than one-half the time, as it usually does, the volume of air required will be proportionately less, and vice versa. The table is computed for maximum loads, which also in practice may vary widely. From the intermittent character of the work of a hoisting engine, the parts are able to resume their normal temperature between the hoists, and there is little probability of the annoyance of freezing up the exhaust passages.

Table of the Volume of Free Air Required for Operating Hoisting Engines

The Air Compressed to 60 Pounds Gauge Pressure.

Diameter of Cylinder, Inches	Stroke, Inches	Revolutions per Minute	Normal Horse-Power	Actual Horse-Power	Weight Lifted Single Rope	Cubic Feet of Free Air Required
SINGLE CYLINDER HOISTING ENGINE						
5	6	200	3	5.9	600	75
5	8	160	4	6.3	1000	80
6¼	8	160	6	9.9	1500	125
7	10	125	10	12.1	2000	151
8¼	10	125	15	16.8	3000	170
8½	12	110	20	18.9	5000	238
10	12	110	25	26.2	6000	330
DOUBLE CYLINDER HOISTING ENGINE						
5	6	200	6	11.8	1000	150
5	8	160	8	12.6	1650	160
6¼	8	160	12	19.8	2500	250
7	10	125	20	24.2	3500	302
8¼	10	125	30	33.6	6000	340
8½	12	110	40	37.8	8000	476
10	12	110	50	52.4	10000	660
12¼	15	100	75	89.2	.....	1125
14	18	90	100	125	.....	1587

Air Used in Cubic Feet Free Air Per Minute Per Indicated Horse-power in Motors (Without Reheating)

Point of Cut-off	Gauge Pressure in Pounds										
	30	40	50	60	70	80	90	100	110	125	150
1	23.3	21.3	20.2	19.4	18.8	18.42	18.10	17.8	17.62	17.40	17.05
¾	18.7	17.1	16.1	15.47	15.0	14.6	14.35	14.15	13.98	13.78	13.50
⅔	17.85	16.2	15.2	14.50	14.2	13.75	13.47	13.28	13.08	12.90	12.60
½	16.4	14.5	13.5	12.8	12.3	11.93	11.7	11.48	11.30	11.10	10.85
⅓	17.5	15.2	12.9	11.85	11.26	10.8	10.5	10.21	10.02	9.78	9.50
¼	20.6	15.6	13.4	13.3	11.40	10.72	10.31	10.0	9.75	9.42	9.10

As will be seen from the table, the only data required are the gauge pressure and point of cut-off; given those two items, we find from the table the free air required per indicated horsepower, and it will only be necessary to multiply this amount by the total indicated horse power of the motor, to determine the total quantity of free air required, and consequently the necessary size of an air compressor to furnish the required amount of air.

These figures do not take account of clearance, but it will be an easy matter to add the per cent of clearance after having determined the total amount of free air required.

It will also be noticed that the free air consumption is based upon the use of cold air, i. e., initial temperature of air at 60 degrees Fahr. In case reheating is resorted to, there will be a corresponding decrease in the amount used, dependent upon the temperature of air on admission to motor, and will be proportional to the ratio of $\frac{T_2}{T_3}$ where $T_2 = 460$ plus 60=520 degrees Fahr. absolute temperature and T_3 =460 plus temperature of air at admission to motor.

Thus, if the air is reheated to 300 degrees Fahr., the quantity in the table will have to be multiplied by

$$\frac{460+60}{460+300} = \frac{520}{760} = .684$$

A further use of this table is to find the most economical point of cut-off for gauge pressures from 30 pounds to 150 pounds per square inch. This fact is apparent from a study of each vertical column: thus, at 60 pounds pressure, the lowest consumption of free air per indicated horse power is at 1-3 cut-off, while at 40 pounds pressure the most economical cut-off will be ½.

Volume of Air and Pressure Required to Drive Direct Acting Steam Pumps

From Hiscox's "Compressed Air"

Head of Water in Feet	Gauge Pressure in Pounds per Square Inch — Ratio of Cylinder Diameters							Cubic Feet of Free Air per Minute to Lift One Gallon of Water — Ratio of Cylinder Diameters						
	1 to 1	1¼ to 1	1½ to 1	1¾ to 1	2 to 1	2½ to 1	3 to 1	1 to 1	1¼ to 1	1½ to 1	1¾ to 1	2 to 1	2½ to 1	3 to 1
10	6							.22						
20	11	7						.28	.37					
30	16	10	7					.33	.42	.53				
40	21	13	9	7				.38	.47	.58	.72			
50	26	17	12	9	7			.44	.53	.65	.79	.94		
60	31	20	14	10	8			.49	.58	70	.82	.99		
70	36	23	16	12	9			.54	.63	.75	.88	1.03		
80	42	26	18	14	11			.61	.68	.79	.95	1.11		
90	47	30	21	15	12			.66	.75	.87	.98	1.15		
100	52	34	23	17	13			.72	.82	.91	1.05	1.20		
125	65	42	29	21	16	10		.86	.95	1.06	1.18	1.33	1.67	
150	78	50	35	25	20	13	9	1.00	1.08	1.20	1.31	1.50	1.88	2.31
175	90	58	40	30	23	15	10	1.12	1.22	1.32	1.47	1.63	2.00	2.40
200	105	67	46	34	26	17	12	1.28	1.37	1.47	1.60	1.75	2.14	2.60
250	....	83	58	42	33	21	15	....	1.64	1.75	1.86	2.06	2.41	2.89
300	....	100	68	50	39	25	17	....	1.92	2.00	2.12	2.31	2.68	3.08
350	....	....	80	58	45	29	20	....	....	2.28	2.39	2.57	2.95	3.37
400	....	....	92	67	52	33	23	....	....	2.57	2.68	2.87	3.22	3.66
450	....	....	105	75	58	37	26	....	....	2.88	2.94	3.13	3.48	3.95
500	....	....	....	85	65	42	29	....	....	....	3.27	3.42	3.82	4.24
600	....	....	....	100	78	50	35	....	....	....	3.76	4.00	4.35	4.80
700	....	....	....	....	92	60	42	....	....	....	....	4.58	5.00	5.50
800	....	....	....	....	105	67	47	....	....	....	....	5.15	5.50	5.96
900	....	....	....	....	....	75	52	....	....	....	....	....	6.00	6.45
1000	....	....	....	....	....	85	58	....	....	....	....	....	6.70	7.00

To find the quantity of free air required per minute, in a direct acting steam pump, to raise a given number of gallons of water through a given head, divide the diameter of the air cylinder by the diameter of the water cylinder, and under the heading of this ratio in above table, and to the right of the given head or lift, find the cubic feet of free air per gallon required per minute; this constant, multiplied by the total number of gallons to be lifted, will give the quantity of free air required. The gauge pressure for the corresponding conditions may be found in a similar manner under the heading of gauge pressures.

In the foregoing table of pressures an allowance of 15 per cent has been made for pump friction, and in the table of volumes 15 per cent has also been allowed for clearance losses and leakage. If the air is reheated before admission to air cylinder, the quantity may be reduced in proportion to the ratio of absolute temperatures. For compound pumps the consumption may be assumed at 75 per cent of the best results of the above table.

Volume of Air (Cubic Feet of Free Air per Minute) that will Flow through Round Holes in a Receiver into the Atmosphere

Diameter of Orifice, Inches	GAUGE PRESSURES IN RECEIVER—POUNDS																		
	2	5	10	15	20	25	30	35	40	45	50	60	70	80	90	100	125	150	200
1/64	.038	.0597	.0842	.103	.119	.133	.156	.173	.19	.208	.225	.26	.295	.33	.364	.40	.486	.57	.76
1/32	.153	.242	.342	.418	.485	.54	.632	.71	.77	.843	.914	1.05	1.19	1.33	1.47	1.61	1.97	2.33	3.07
3/64	.342	.545	.77	.94	.107	1.21	1.4	1.56	1.71	1.91	2.05	2.35	2.68	2.97	3.28	3.66	4.12	5.20	6.9
1/16	.647	.965	1.36	1.67	1.93	2.16	2.52	2.80	3.07	3.36	3.64	4.2	4.76	5.32	5.87	6.45	7.85	9.20	22.2
3/32	1.37	2.18	3.06	3.75	4.28	4.84	5.6	6.24	6.84	7.6	8.2	9.4	10.7	11.88	13.1	14.5	17.5	20.8	27.5
1/8	2.435	3.86	5.45	6.65	7.7	8.6	10	11.2	12.27	13.4	14.5	10.8	19	21.2	23.5	25.8	31.4	36.7	48.7
3/16	5.45	8.72	12.3	15	17.1	19.4	22.4	25	27.4	30.4	32.8	37.5	42.9	47.5	52.4	58	70	83.2	110
1/4	9.74	15.4	21.8	26.7	30.8	34.5	40	44.7	49.09	53.8	58.2	67	76	85	94	103.2	125.5	147.	195.8
3/8	21.95	34.6	49	60	69	77	90	100	110.5	121	130	151	171	191	211	231	282	330	440
1/2	39	61.6	87	107	123	138	161	179	196.4	215	232	268	304	340	376	412	502	588	782
5/8	61	96.5	136	167	193	216	252	280	306.8	336	364	420	476	532	587	645	785	920	1220
3/4	87.6	133	196	240	277	310	362	400	442	482	521	604	685	765	843	925	1127	1322	1760
7/8	119.5	189	267	326	378	422	493	550	601	658	710	622	930	1004	1145	1260	1161	1804	
1	156	247	350	427	494	550	645	715	786	860	930	1070	1215	1360	1500	1648	2000	2350	
1 1/8	196	310	440	538	620	692	812	900	987	1082	1170	1350	1530	1710	1890				
1 1/4	242	384	543	665	770	860	1000	1120	1230	1345	1455	1680	1900	2130	2350				
1 3/8	291	464	660	803	929	1040	1200	1350	1480	1630	1760								
1 1/2	350	550	780	960	1102	1240	1435	1600	1770	1930	2085								
1 3/4	473	752	1068	1304	1505	1680	1960												
2	625	985	1395	1700	1967	2200	2586												

Cubic Feet of Free Air Required to Run One Drill of Size and at the Pressure Stated Below

Gauge Pressure	CYLINDER DIAMETER OF DRILL IN INCHES												
	2	2¼	2½	2¾	3	3⅛	3 3/16	3¼	3½	3⅝	4¾	5	5½
60	50	60	68	82	90	95	97	100	108	113	130	150	164
70	56	68	77	93	102	108	110	113	124	129	147	170	181
80	63	76	86	104	114	120	123	127	131	143	164	190	207
90	70	84	95	115	126	133	136	141	152	159	182	210	230
100	77	92	104	126	138	146	149	154	166	174	199	240	252

Multipliers to Determine Capacity of Compressor Required to Operate from 1 to 70 Rock Drills at Altitudes Compared with Sea Level

Altitude Above Sea Level	NUMBER OF DRILLS																		
	1	2	3	4	5	6	7	8	9	10	12	15	20	25	30	40	50	60	70
	MULTIPLIERS																		
0	1.	1.8	2.7	3.4	4.1	4.8	5.4	6.	6.5	7.1	8.1	9.5	11.7	13.7	15.8	21.4	25.5	29.4	33.2
1000	1.03	1.85	2.78	3.5	4.22	4.94	5.56	6.18	6.69	7.3	8.34	9.78	12.05	14.1	16.3	22.	26.26	30.3	34.2
2000	1.07	1.92	2.89	3.64	4.39	5.14	5.78	6.42	6.95	7.6	8.67	10.17	12.52	14.66	16.9	22.9	27.28	31.46	35.52
3000	1.10	1.98	2.97	3.74	4.51	5.28	5.94	6.6	7.15	7.81	8.91	10.45	12.87	15.07	17.38	23.54	28 05	32.34	36.52
4000	1.14	2.05	3.08	3.88	4.67	5.47	6.15	6.84	7.41	8.09	9.23	10.83	13.34	15.62	18.01	24.4	29.07	33.52	37.8
5000	1.17	2.10	3.16	3.98	4.8	5.62	6.32	7.02	7.61	8.31	9.48	11.12	13.69	16.03	18.49	25.04	29.84	34.4	38.84
6000	1.20	2.16	3.24	4.08	4.9	5.76	6.48	7.2	7.8	8.52	9.72	11.4	14.04	16.44	18.96	25.68	30.6	35.4	39.84
7000	1.23	2.21	3.32	4.18	5.04	5.9	6.64	7.38	7.99	8 73	9.96	11.68	14.39	16.85	19.43	26.32	31.36	36.16	40.84
8000	1.26	2.27	3.40	4.28	5.17	6.05	6.8	7.56	8.19	8.95	10.21	11.97	14.74	17.26	19.9	26.96	32.13	37.04	41.83
9000	1.29	2.32	3.48	4.39	5.29	6.19	6.96	7.74	8.38	9.16	10.45	12.26	15.09	17.67	20.38	27.6	32.9	37.92	42.83
10000	1.32	2.38	3.56	4.49	5.41	6.34	7.13	7.92	8.58	9.37	10.69	12.54	15.44	18.08	20.86	28.25	33.66	38.8	43.82
12000	1.37	2.47	3.7	4.66	5.62	6.57	7.4	8.22	8.9	9.73	11.1	13.02	16.03	18.77	21.64	29.32	34.94	40.28	45.48
15000	1.43	2.57	3.86	4.86	5.86	6.86	7.72	8.58	9.3	10.15	11.58	13.58	16.73	19.59	22.59	30.6	36.46	42.04	47.47

Contents of Cylinder in Cubic Feet for Each Foot in Length

Diameter in Inches	Cubic Contents	Diameter in Inches	Cubic Contents	Diameter in Inches	Cubic Contents	Diameter in Inches	Cubic Contents	Diameter in Inches	Cubic Contents
1	.0055	6	.1963	11	.6600	20	2.182	36	7.069
1¼	.0085	6¼	.2130	11¼	.6903	20½	2.292	37	7.468
1½	.0123	6½	.2305	11½	.7213	21	2.405	38	7.886
1¾	.0168	6¾	.2485	11¾	.7530	21½	2.521	39	8.296
2	.0218	7	.2673	12	.7854	22	2.640	40	8.728
2¼	.0276	7¼	.2868	12½	.8523	22½	2.761	41	9.168
2½	.0341	7½	.3068	13	.9218	23	2.885	42	9.620
2¾	.0413	7¾	.3275	13½	.9940	23½	3.012	43	10.084
3	.0491	8	.3490	14	1.069	24	3.142	44	10.560
3¼	.0576	8¼	.3713	14½	1.147	25	3.409	45	11.044
3½	.0668	8½	.3940	15	1.227	26	3.687	46	11.540
3¾	.0767	8¾	.4175	15½	1.310	27	3.976	47	12.048
4	.0873	9	.4418	16	1.396	28	4.276	48	12.566
4¼	.0985	9¼	.4668	16½	1.485	29	4.587		
4½	.1105	9½	.4923	17	1.576	30	4.909		
4¾	.1231	9¾	.5185	17½	1.670	31	5.241		
5	.1364	10	.5455	18	1.767	32	5.585		
5¼	.1503	10¼	.5730	18½	1.867	33	5.940		
5½	.1650	10½	.6013	19	1.969	34	6.305		
5¾	.1803	10¾	.6303	19½	2.074	35	6.681		

To ascertain the piston displacement of an air compressor, multiply the cubic contents stated opposite its diameter (second column) by the piston travel in feet per minute. The result will be the theoretical capacity of each double-acting cylinder, or one-half of the result will be the theoretical capacity of a single-acting cylinder.

Table of Branch Pipes

Relative Carrying Capacity of Pipes for Air.

Diam. of Pipe	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	10	12
1	1.00	.52	.327	.15	.084	.05									
1¼	1 90	1.00	.614	.28	.16	.10	.066								
1½	3.05	1.60	1.00	.46	.256	.16	.106	.075							
2	6.55	3.45	2.14	1.00	.56	.34	.23	.160	.12						
2½	11.8	6.25	3.88	1.81	1.00	.614	.41	.29	.216	.163					
3	19.0	12.0	6 32	2 95	1.63	1.00	.67	.47	.35	.268	.165				
3½	.....	15.2	9.45	4.3	2.43	1.50	1.00	.71	.52	.400	.246	.166			
4	.....	21.6	13.4	6.25	3.46	2.10	1.42	1.00	.75	.56	.352	.237	.169		
4½	.....	.....	18.0	8.30	4.65	2.85	1.90	1.35	1.00	.76	.475	.32	.227	.128	
5	.....	.....	.....	.....	6.14	3.77	2.60	1.78	1.32	1.00	.625	.42	.30	.169	.10
6	.....	.....	.....	.....	.....	6.05	4.00	2.85	2.15	1.60	1.00	.675	.48	.27	.17
7	.....	.....	.....	.....	.....	.....	6.00	4.20	3.16	2.37	1.48	1.00	.71	.40	.25
8	.....	.....	.....	.....	.....	.....	.....	6.00	4.40	3.25	2.10	1.40	1.00	.56	.35
10	.....	.....	.....	.....	.....	.....	.....	.....	7 85	5.90	3.70	2 50	1.77	1.00	.63
12	.....	.....	.....	.....	.....	.....	.....	.....	.....	9 40	5.90	3.95	2.80	1.60	1.00

Table Showing Loss in Pressure, in Pounds, Due to Friction in Pipes 100 Feet in Length

Gauge Pressure at Entrance to Pipe, 75 Pounds

Diam. Pipe	CUBIC FEET FREE AIR DELIVERED PER MINUTE																													
	25	50	75	100	125	150	200	250	300	350	400	500	600	700	800	900	1000	1200	1500	1800	2000	2500	3000	3500	4000	5000	6000	7000	8000	9000	10000
1	.38	1.51	3 4	6.05																											
1¼	.11	.42	.95	1.69																											
1½	....	.16	.37	.65	1.02	1.46	2.6																								
2	....	.04	.08	.14	.22	.32	.56	.88	1.26	1.72	2.25																				
2½	....	....	....	.04	.07	.1	.17	.27	.39	.53	.69	1.08	1.55	2.11	2.76																
3	....	....	....	....	....	.04	.06	.1	.15	.2	.26	.4	.58	.79	1.03	1.3	1.61	2.32													
3½	....	....	....	....	....	....	....	....	.07	.09	.12	.18	.26	.36	.47	.59	.73	1.05	1.65												
4	....	....	....	....	....	....	....	....	....	....	.06	.09	.13	.18	.23	.29	.36	.52	.81	1.17	1.44	2.26									
4½	....	....	....	....	....	....	....	....	....	....	....	.05	.07	.1	.13	.16	.2	.29	.45	.65	.8	1.25	1.81								
5	....	....	....	....	....	....	....	....	....	....	....	....	.04	.05	.07	.09	.11	.16	.25	.36	.44	.69	1.								
6	....	....	....	....	....	....	....	....	....	....	....	....	....	....	.03	.04	.05	.06	.1	.15	.18	.28	.4	.55	.72						
7	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	.02	.03	.05	.07	.08	.13	.18	.25	.32	.5	.73				
8	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	.02	03	.04	.06	.09	.13	.16	.26	.37	.5	.66		
10	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	.01	.02	.03	.04	.05	.08	.12	.16	.21	.27	.33
12	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	.01	.02	.02	.03	.05	.07	.09	.11	.14

The loss of pressure from friction in transmitting air through pipe is very slight and requires consideration in figuring the efficiency of a compressed air installation. It should be remembered that a reduction in pressure due to friction does not indicate an absolute loss of air, as the **volume** delivered will show an equivalent increase Sharp turns or abrupt angles in the pipe should be avoided, if possible, and it is desirable to test the line occasionally by allowing the full pressure of air to remain in the pipe for a short period of time. Any leakage will thus become evident through a reduction in gauge pressure and should be immediately located and stopped, as the constant escape of air from even the smallest opening will result in a most serious waste of power.

The above table will prove of assistance in determining proper diameters of compressed air transmission pipes.

Table Showing Loss in Pressure, in Pounds, Due to Friction in Pipes 100 Feet in Length

Gauge Pressure at Entrance to Pipe, 90 Pounds

Diam. Pipe	CUBIC FEET FREE AIR DELIVERED PER MINUTE																													
	25	50	75	100	125	150	200	250	300	350	400	500	600	700	800	900	1000	1200	1500	1800	2000	2500	3000	3500	4000	5000	6000	7000	8000	9000	10000
1	.33	1.3	2.92	5 19																											
1¼	.09	.36	.81	1 44																											
1½	....	.14	.31	.56	.87	1.25	2.23																								
2	....	.03	.07	.12	.19	.27	.48	.76	1 09	1.48	1 94	....	....	....	....																
2½	....	....	....	.04	.06	.08	.15	.23	.33	.45	.59	.92	1.33	1.81	2.36																
3	....	....	....	....	....	.03	.06	.09	.13	.17	.22	.35	.5	.68	.89	1.12	1.39	2.													
3½	....	....	....	....	....	....	....	....	.06	.08	.1	.16	.23	.3	.4	.51	.63	.9	1.41												
4	....	....	....	....	....	....	....	....	....	....	.05	.08	.11	.15	.2	.25	31	.45	.7	1.	1.24	1.93									
4½	....	....	....	....	....	....	....	....	....	....	....	.04	.06	.08	.11	.14	.17	.25	.39	.56	.69	1.07	1.54								
5	....	....	....	....	....	....	....	....	....	....	....	....	.04	.05	.06	.08	.1	.14	.22	.32	.39	.61	88								
6	....	....	....	....	....	....	....	....	....	....	....	....	....	....	.02	.03	.04	.06	.09	.12	.15	.24	.35	.47	.61						
7	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	.02	.03	.04	.06	.07	.11	.16	.12	28	.44	.63				
8	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	.02	.03	.04	.06	.08	.11	.14	.22	.32	.43	.57		
10	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	.01	.02	.03	.03	.05	.07	.1	.14	.18	.23	.28
12	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	.01	.01	.02	.03	.04	.05	.07	.09	.11

Table Showing Loss in Pressure, in Pounds, Due to Friction in Pipes 100 Feet in Length

Gauge Pressure at Entrance to Pipe, 100 Pounds

Diam. Pipe	CUBIC FEET FREE AIR DELIVERED PER MINUTE																													
	25	50	75	100	125	150	200	250	300	350	400	500	600	700	800	900	1000	1200	1500	1800	2000	2500	3000	3500	4000	5000	6000	7000	8000	9000	10000
1	.3	1.18	2.66	4.75																											
1¼	.08	.33	.73	1.3																											
1½	....	.13	.29	.6	.8	1.15	2.04																								
2	....	.03	.07	.12	.18	.26	.47	.73	1.05	1.43	1.87																				
2½	....	....	....	.03	.05	.08	.14	.21	.31	.42	.54	.85	1.22	1.66	2.17																
3	....	....	....	....	....	.03	.05	.08	.11	.15	2	.31	.45	.61	.8	1.01	1.25	1.8													
3½	....	....	....	....	....	....	....	....	.05	.07	.09	.14	.21	.28	.37	.46	.57	.82	1.28												
4	....	....	....	....	....	....	....	....	....	....	.05	.07	.1	.14	.18	.23	.28	.41	.64	.92	1.13	1.77									
4½	....	....	....	....	....	....	....	....	....	....	....	.04	.06	.08	.1	.13	.16	.23	.35	.51	.63	.98	1.41								
5	....	....	....	....	....	....	....	....	....	....	....	....	.03	.04	.06	.07	.09	.13	.2	.29	.36	.56	.81								
6	....	....	....	....	....	....	....	....	....	....	....	....	....	....	.02	.03	.04	.05	.08	.11	.14	.22	.32	.43	.56						
7	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	.02	.02	.04	.05	.06	.1	.14	.2	.25	.4	.57				
8	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	.02	.03	.03	.05	.07	.1	.13	.2	.29	.39	.51		
10	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	.01	.02	.02	.03	.04	.06	.09	.13	.16	.21	.26
12	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	.01	.01	.02	.02	.04	.05	.06	.08	.1

Globe Valves, Tees and Elbows

The reduction of pressure produced by globe valves is the same as that caused by the following additional lengths of straight pipe, as calculated by the formula:

$$\text{Additional length of pipe} = \frac{114 \times \text{diameter of pipe}}{1 + (3.6 = \text{diameter})}$$

Diameter of pipe	1	1½	2	2½	3	3½	4	5	6 ins.
Additional length	2	4	7	10	13	16	20	28	36 ft.
	7	8	10	12	15	18	20	22	24 ins.
	44	53	70	88	115	143	162	181	200 ft.

The reduction of pressure produced by elbows and tees is equal to two-thirds of that caused by globe valves. The following are the additional lengths of straight pipe to be taken into account for elbows and tees. For globe valves, multiply by ⅔:

Diameter of pipe	1	1½	2	2½	3	3½	4	5	6 ins.
Additional length	2	3	5	7	9	11	13	19	24 ft.
	7	8	10	12	15	18	20	22	24 ins.
	30	35	47	59	77	96	108	120	134 ft.

These additional lengths of pipe for globe valves, elbows and tees must be added in each case to the actual length of straight pipe. Thus a 6-inch pipe, 500 feet long, with 1 globe valve, 2 elbows and 3 tees, would be equivalent to a straight pipe $500 + 36 + (2 \times 24) + (3 \times 24) = 656$ feet long.

Table Giving the Theoretical Volume of Equivalent Free Air, in Cubic Feet

That will flow per minute at various pressures through straight pipes of different diameters, each 100 feet long, without any reduction of pressure.

Initial and Terminal Gauge Pressure, Lbs.	NOMINAL DIAMETERS OF PIPES IN INCHES																		Initial and Terminal Gauge Pressure, Lbs.
	1	1½	2	2½	3	3½	4	5	6	7	8	10	12	15	18	20	22	24	
10	24.05	75.67	153.0	244.7	433.3	636.4	883.8	1585	2542	3701	5134	9212	14530	22530	36280	48180	61870	77880	10
20	28.48	89.54	177.9	288.8	512.9	753.0	1045	1871	3006	4381	6076	10900	17210	26640	42940	57000	73220	92150	20
30	32.35	101.6	202.0	329.1	582.5	855.0	1187	2124	3415	4975	6900	12380	19540	30270	48750	64760	83160	104630	30
40	35.82	112.5	223.5	364.2	644.7	946.4	1314	2352	3779	5507	7635	13700	21620	33500	53960	71640	92010	115800	40
50	38.96	122.4	243.4	396.3	701.5	1030.	1430	2558	4114	5993	8312	14910	23520	36450	58730	74300	100150	126030	50
60	41.83	131.4	261.1	425.4	752.9	1105.	1535	2747	4416	6433	8920	16000	25260	39120	63040	83660	107470	135240	60
70	44.53	139.9	278.0	452.9	801.8	1176	1634	2925	4701	6848	9499	17030	26900	41660	67110	89100	114420	144000	70
80	47.08	147.9	294.0	478.8	847.6	1244	1728	3091	4971	7240	10040	18000	28430	44050	70960	94180	120970	152230	80
90	49.54	155.6	309.3	503.8	891.8	1307	1817	3253	5230	7619	10560	18940	29920	46340	74660	99100	127300	160190	90
100	51.88	163.0	324.0	527.5	933.8	1370	1904	3407	5477	7979	11050	19850	31330	48530	78190	103770	133300	167720	100
110	54.10	169.9	337.8	550.1	973.9	1429	1985	3552	5712	8320	11530	20690	32670	50610	81540	108220	139000	174920	110
125	57.15	179.5	356.8	581.3	1028	1510	2097	3754	6034	8789	12180	21860	34520	53470	86140	114430	146850	184800	125
150	62.10	195.1	387.8	631.7	1117	1641	2280	4080	6558	9553	13240	23760	37520	58120	93610	124280	159600	.....	150
175	66.71	209.6	416.5	678.4	1201	1762	2448	4382	7044	10250	14220	25520	40300	62420	100520	133460	.....	.....	175
200	70.93	222.9	443.0	721.4	1277	1874	2603	4659	7489	10900	15120	27140	42850	66370	106900	141910	.....	.....	200
250	78.70	247.3	491.4	800.4	1416	2079	2889	5170	8309	12100	16780	30110	47550	73640	118620	.....	.....	.....	250
300	85.85	269.9	536.3	873.4	1546	2269	3152	5642	9063	13210	18320	32870	51890	80380	.....	.....	.....	.....	300
400	98.55	309.9	615.7	1002	1775	2606	3619	6477	10400	15160	21020	37730	59580	92270	.....	.....	.....	.....	400
500	109.7	345.1	685.3	1116	1976	2902	4031	7215	11590	16890	23420	42020	66340	102750	.....	.....	.....	.....	500
600	119.9	377.3	749.5	1220	2160	3172	4405	7882	12670	18460	25600	45940	72520	.....	.....	.....	.....	.....	600
700	129.3	406.6	807.6	1315	2328	3419	4748	8496	13650	19890	27590	49500	78150	.....	.....	.....	.....	.....	700
800	138.1	434.1	862.0	1404	2486	3650	5070	9072	14580	21240	29450	52840	.....	.....	.....	.....	.....	.....	800
900	146.3	460.0	914.4	1489	2636	3871	5373	9618	15460	22520	31230	56040	.....	.....	.....	.....	.....	.....	900
1000	154.3	485.1	963.5	1569	2778	4079	5664	10130	16290	23730	32920	59060	.....	.....	.....	.....	.....	.....	1000
1100	161.3	507.1	1008	1641	2906	4264	5924	10590	17040	24820	34430	.....	.....	.....	.....	.....	.....	.....	1100
1200	168.6	530.3	1053	1716	3037	4457	6193	11080	17810	25940	35980	.....	.....	.....	.....	.....	.....	.....	1200
1300	175.7	552.2	1097	1787	3163	4643	6448	11540	18550	27030	.....	.....	.....	.....	.....	.....	.....	.....	1300
1400	182.2	572.4	1137	1853	3280	4814	6686	11970	19230	28020	.....	.....	.....	.....	.....	.....	.....	.....	1400
1500	188.2	591.5	1175	1914	3389	4973	6908	12360	19870	.....	.....	.....	.....	.....	.....	.....	.....	.....	1500

Weight of Air at Various Pressures and Temperatures

Based on an Atmospheric Pressure of 14.7 Pounds Absolute at Sea Level

Temp. of Air Deg. Fahr.	GAUGE PRESSURE, POUNDS																					
	0	5	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	175	200	225	250	300
	WEIGHT IN POUNDS PER CUBIC FOOT																					
—20	.0900	.1205	.1515	.2125	.2744	.3360	.3970	.4580	.5190	.5800	.6410	.702	.7635	.825	.886	.948	1.010	1.165	1.318	1.465	1.625	1.930
—10	.0882	.1184	.1485	.2090	.2685	.3283	.3880	.4478	.5076	.5674	.6272	.687	.747	.807	.868	.928	.989	1.139	1.288	1.438	1.588	1.890
0	.0864	.1160	.1455	.2040	.2630	.3215	.3800	.4385	.4970	.5555	.6140	.672	.731	.790	.849	.908	.968	1.114	1.260	1.406	1.553	1.850
10	.0846	.1136	.1425	.1995	.2568	.3145	.3720	.4292	.4863	.5433	.6006	.658	.716	.774	.832	.889	.947	1.090	1.233	1.376	1.520	1.810
20	.0828	.1112	.1395	.1955	.2516	.3071	.3645	.4205	.4770	.5330	.5890	.645	.701	.757	.813	.869	.927	1.067	1.208	1.348	1.489	1.770
30	.0811	.1088	.1366	.1916	.2465	.3015	.3570	.4121	.4672	.5221	.5771	.632	.687	.742	.797	.852	.908	1.046	1.184	1.322	1.460	1.735
40	.0795	.1067	.1338	.1876	.2415	.2954	.3503	.4038	.4576	.5114	.5652	.619	.673	.727	.781	.835	.890	1.025	1.161	1.296	1.431	1.701
50	.0780	.1045	.1310	.1839	.2367	.2905	.3432	.3960	.4487	.5014	.5541	.607	.660	.713	.766	.819	.873	1.006	1.139	1.271	1.403	1.668
60	.0764	.1025	.1283	.1803	.2323	.2840	.3562	.3882	.4402	.4927	.5447	.596	.649	.700	.752	.804	.856	.988	1.116	1.245	1.376	1.636
70	.0750	.1005	.1260	.1770	.2280	.2791	.3302	.3808	.4316	.4824	.5332	.584	.635	.686	.737	.788	.839	.967	1.095	1.223	1.350	1.604
80	.0736	.0988	.1239	.1738	.2237	.2739	.3242	.3738	.4234	.4729	.5224	.572	.622	.673	.723	.774	.824	.949	1.074	1.199	1.325	1.573
90	.0723	.0970	.1218	.1707	.2195	.2688	.3182	.3670	.4154	.4639	.5122	.561	.611	.660	.709	.759	.809	.932	1.054	1.177	1.300	1.544
100	.0710	.0954	.1197	.1676	.2155	.2638	.3122	.3602	.4079	.4555	.5033	.551	.599	.648	.696	.745	.794	.914	1.035	1.155	1.276	1.517
110	.0698	.0937	.1176	.1645	.2115	.2593	.3070	.3542	.4011	.4481	.4950	.542	.589	.637	.685	.732	.780	.899	1.017	1.135	1.254	1.491
120	.0686	.0921	.1155	.1618	.2080	.2549	.3018	.3481	.3944	.4403	.4866	.533	.579	.626	.673	.720	.767	.884	1.001	1.118	1.234	1.465
130	.0674	.0905	.1135	.1590	.2045	.2505	.2966	.3446	.3924	.4296	.4770	.524	.570	.616	.662	.708	.754	.869	.984	1.099	1.214	1.440
140	.0663	.0889	.1115	.1565	.2015	.2465	.2915	.3364	.3813	.4262	.4711	.516	.561	.606	.651	.696	.742	.855	.968	1.081	1.194	1.416
150	.0652	.0874	.1096	.1541	.1985	.2425	.2865	.3308	.3751	.4193	.4636	.508	.552	.596	.640	.685	.730	.841	.953	1.064	1.175	1.392
175	.0626	.0840	.1054	.1482	.1910	.2335	.2755	.3181	.3607	.4033	.4450	.488	.531	.573	.616	.658	.701	.808	.914	1.021	1.128	1.337
200	.0603	.0809	.1014	.1427	.1840	.2248	.2655	.3054	.3473	.3882	.4291	.470	.511	.552	.592	.633	.674	.776	.879	.982	1.084	1.287
225	.0581	.0779	.0976	.1373	.1770	.2163	.2555	.2949	.3344	.3738	.4129	.452	.491	.531	.570	.609	.649	.747	.846	.944	1.043	1.240
250	.0560	.0751	.0941	.1323	.1705	.2085	.2466	.2845	.3223	.3602	.3981	.436	.474	.513	.551	.589	.627	.722	.817	.912	1.007	1.197
275	.0541	.0726	.0910	.1278	.1645	.2011	.2378	.2745	.3111	.3478	.3844	.421	.458	.494	.531	.568	.605	.697	.789	.881	.972	1.155
300	.0523	.0707	.0881	.1237	.1592	.1945	.2300	.2654	.3008	.3362	.3716	.407	.442	.478	.513	.549	.585	.673	.762	.852	.940	1.118
350	.0491	.0658	.0825	.1160	.1495	.1828	.2160	.2492	.2824	.3156	.3488	.382	.415	.449	.482	.516	.549	.632	.715	.799	.883	1.048
400	.0463	.0621	.0779	.1090	.1405	.1720	.2035	.2348	.2661	.2974	.3287	.360	.391	.423	.454	.486	.517	.596	.674	.753	.831	.987
450	.0437	.0586	.0735	.1033	.1330	.1628	.1925	.2220	.2515	.2810	.3105	.340	.369	.399	.429	.458	.488	.562	.637	.711	.786	.934
500	.0414	.0555	.0696	.0978	.1260	.1540	.1820	.2100	.2380	.2660	.2940	.322	.351	.379	.407	.435	.463	.534	.604	.675	.746	.885
550	.0394	.0528	.0661	.0930	.1198	.1464	.1730	.1996	.2262	.2528	.2794	.306	.333	.359	.386	.413	.440	.507	.573	.641	.709	.841
600	.0376	.0504	.0631	.0885	.1140	.1395	.1650	.1904	.2158	.2412	.2668	.292	.317	.343	.368	.393	.419	.483	.547	.611	.675	.801

Cylinder Piston Displacements

In Cubic Feet Per Revolution — Double Acting

Diam. in Inches	STROKE IN INCHES														
	6	7	8	10	12	14	16	18	20	22	24	30	36	42	48
6	.196	.229	.261	.327	.392	.458	.523	.....	.....	.....	.....	.....	.....	.....	.....
7	.267	.311	.356	.445	.534	.624	.712	.....	.....	.....	.....	.....	.....	.....	.....
8	.349	.407	.465	.581	.698	.814	.930	1.04	1.16	1.27	1.39	1.74	.....	.....	.....
9	.442	.515	.588	.736	.883	1.030	1.177	1.32	1.47	1.61	1.76	2.20	.....	.....	.....
10	.545	.636	.727	.909	1.090	1.272	1.454	1.63	1.81	2.00	2.18	2.72	3.27	.....	.....
11	.659	.769	8.79	1.099	1.319	1.539	1.759	1.97	2.19	2.41	2.63	3.29	3.95	.....	.....
12	.785	.916	1.047	1.308	1.571	1.832	2.094	2.35	2.61	2.87	3.14	3.92	4.71	5.49	.....
13	.....	1.075	1.228	1.536	1.843	2.150	2.457	2.76	3.07	3.37	3.68	4.60	5.53	6.45	.....
14	.....	1.247	1.425	1.781	2.137	2.494	2.850	3.20	3.56	3.91	4.27	5.34	6.41	7.48	8.55
15	.....	.....	1.636	2.045	2.454	2.863	3.272	3.68	4.09	4.49	4.90	6.13	7.36	8.58	9.81
16	.....	.....	1.861	2.327	2.792	3.257	3.723	4.18	4.65	5.12	5.58	6.98	8.37	9.77	11.16
17	.....	.....	.....	2.629	3.152	3.677	4.203	4.72	5.25	5.77	6.30	7.88	9.45	11.03	12.60
18	.....	.....	.....	2.945	3.534	4.123	4.712	5.30	5.89	6.47	7.06	8.83	10.60	12.36	14.13
19	.....	.....	.....	.....	3.937	4.593	5.250	5.90	6.56	7.21	7.87	9.84	11.81	13.78	15.75
20	.....	.....	.....	.....	4.363	5.090	5.817	6.54	7.27	7.99	8.72	10.90	13.08	15.27	17.45
21	.....	.....	.....	.....	.....	5.612	6.413	7.21	8.01	8.81	9.62	12.02	14.43	16.83	19.24
22	.....	.....	.....	.....	.....	6.159	7.037	7.91	8.79	9.67	10.55	13.19	15.83	18.47	21.11
23	.....	.....	.....	.....	.....	.....	7.693	8.65	9.61	10.57	11.54	14.42	17.31	20.19	23.08
24	.....	.....	.....	.....	.....	.....	8.377	9.42	10.47	11.52	12.56	15.70	18.84	21.98	25.12
25	.....	.....	.....	.....	.....	.....	9.000	10.22	11.36	12.49	13.63	17.04	20.44	23.85	27.26
26	.....	.....	.....	.....	.....	.....	9.830	11.06	12.29	13.51	14.74	18.43	22.12	25.80	29.49
27	.....	.....	.....	.....	.....	.....	10.602	11.92	13.25	14.57	15.90	19.88	23.85	27.83	31.80
28	.....	.....	.....	.....	.....	.....	11.402	12.82	14.25	15.67	17.10	21.38	25.65	29.93	34.20
29	.....	.....	.....	.....	.....	.....	.....	13.75	15.28	16.81	18.34	22.93	27.51	32.10	36.68
30	.....	.....	.....	.....	.....	.....	.....	14.72	16.36	17.99	19.63	24.54	29.44	34.35	39.26
31	.....	.....	.....	.....	.....	.....	.....	15.72	17.47	19.21	20.96	26.20	31.44	36.68	41.92
32	.....	.....	.....	.....	.....	.....	.....	16.75	18.61	20.47	22.34	27.92	33.51	39.09	44.68
33	.....	.....	.....	.....	.....	.....	.....	.....	.....	21.77	23.75	29.69	35.63	41.57	47.51
34	.....	.....	.....	.....	.....	.....	.....	.....	.....	23.11	25.22	31.52	37.83	44.13	50.44
35	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	26.72	33.40	40.08	46.76	53.45
36	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	28.27	35.34	42.40	49.47	56.54
37	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	37.33	44.79	52.26	59.72
38	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	39.37	47.25	55.12	63.00
39	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	41.47	49.77	58.06	66.36
40	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	43.63	52.35	61.08	69.80
42	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	67.34	76.96
44	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	73.89	84.46

Air Consumption of Various Types of Tools and Machines

Tools	Size	Air Pressure in in Pounds per Square Inch	Air Consumed, Free Air per Min. (Cubic Feet)
Aerons (Paint Sprays)	Small Hand	90	2–7
Chipping Hammers	Weight Pounds		
	5	90	9
	7	90	12
	8	90	15
	9	90	15½
	10	90	16
	11	90	17
	12	90	18
	13	90	20
	14	90	20
	18	90	22
Foundry Jolting Machines	Platform type	80	Air per ton lifting capacity 30–40
Grinders (Hand)	Weight Pounds		
	17	80	20
	24	80	30
Cylinder Air Hoists Direct Lift	(Cylinder) Diam. inches		Free air in cu. ft. per min. per ft. lift
	6	80	1.22
	8	80	2.24
	10	80	3.29
	12	80	5.06
	14	80	7.13
	17	80	10.10
	19	80	12.50
Cylinder Air Hoists Rope-Geared 2-1	6	80	.61
	8	80	1.12
	10	80	1.65
	12	80	2.53
	14	80	3.57
	17	80	5.05
	19	80	6.25
Air Motor Hoists	Tons Capacity		
	1	80	4
	1½	80	6
	2	80	8
	3	80	9
	4	80	12
	5	80	15
	6	80	19
	8	80	25
	10	80	30

Air Consumption is shown in terms of "Free Air."

Air Consumption of Various Types of Tools and Machines (Continued)

Tools	Size	Air Pressure in pounds per square inch	Air Consumed Cubic Feet Free Air
Compressed Air Jacks	Cylinder Diam. inches		Air per min. per ft. lift
	8	60	1.75
	10	60	2.77
	12	60	3.99
	14	60	5.42
	16	60	6.91
	20	60	11.1
	24	60	15.8
Air Motors	Horsepower		Air per min. (ft.)
	2	80	40–50
	4	80	60–75
	5	80	90–100
	8	80	125
	15	80	240
Molding Machines and Squeezers		80	Cu. ft. air per mold ½–1½
Pile Hammers	Weight Lbs.		Air Per Minute
	145	80	100
	640	80	150
	1500	80	200
	5000	80	350
	7500	80	600
Hand Riveters	Weight Lbs.		Air per minute
	13	90	18
	14 or 15	90	19
	16 or 17	90	20
	19 or 20	90	22
	21 or 22	90	24
	23 or 24	90	25
	25	90	25
Riveting Machines	Reach		Air per rivet
	12 to 30	80	5
	36 to 45	80	6½
Rotary Drills	Weight Lbs.		Air per minute
	10	90	15
	15	90	18
	20	90	20
	30	90	25
	35	90	27
	40	90	30
	45	90	35
	50	90	40
	60	90	43
	70	90	45
	75	90	48

Tools	Size	Air Pressure in pounds per Square Inch	Air Consumed Cubic ft. free air per minute
Rotary Flue Rolling, Reaming and Tapping Machines	30	90	20
	40	90	25
	50	90	30
	55	90	35
	65	90	35
	130	90	45
Sand Blast	Size nozzle, in.		
		80	47
	¼	80	85
	⅜	80	190
	½	80	340
Sand Rammers (hand type)	Weight Lbs.		
	7	80	9
	18	80	15
	24	80	20
	30	80	25
Sand Rammers (Suspended)	Weight Lbs.		
	280	80	50
	325	80	90
	370	80	100
	925	80	130
Stone Cutting Tools	Weight Lbs.		
	1	70	3–6
	2	70	4–7
	3	70	5–8
	4	70	6–9
	5	70	7–10
	9	70	12
Surfacers	Small size Weight approx. 35 pounds	90	30–40
	Large size Weight approx. 65 pounds	90	60–70
Wood Boring Machines	Weight Lbs.		
	15	90	15
	25	90	20
	40	90	30

Centrifugal Pumps

SECTION TWO

Centrifugal Pumps

For many years the centrifugal pump, which is the simplest type of pumping machine, undeveloped and crude in design and construction, was used only for pumping against comparatively low heads, and where economy was of secondary importance. In recent years, however, a great advance has been made in the design and construction of centrifugal pumps; and the higher degree of economy now secured, both at high and low heads, also the use of the turbine pump constructed in multiple stages, have brought the centrifugal pump into general use for a vast variety of purposes. Formerly it was not thought possible to operate centrifugal pumps at heads greater than 20 or 30 feet with any degree of economy, but with the remarkable progress that has been made in both the theory and design, it is now possible to build economical pumps for heads up to 300 feet per stage.

In a centrifugal pump, the mechanical power delivered to the shaft by the prime mover is transmitted to the water by means of a series of radial vanes cast together to form a single element called an impeller, and revolved by the shaft. The water is led to the inner ends of the impeller vanes which pick it up with a rapidly accelerating motion causing it to flow radially between them so that when reaching the outer circumference of the impeller, the water, owing to the velocity and pressure acquired, has absorbed all of the power transmitted to the pump shaft; thus the problem to be solved in impeller design is to obtain the acquired velocity and pressure with a minimum loss in shock and friction. Since the energy of the water on leaving the pump is required to be mostly in the form of a pressure, the next problem is to transform into pressure the kinetic energy of the water due to its velocity on leaving the impeller, and to accomplish it with the least possible loss.

The accomplishment of this is the function of the casing which may be of the volute type, or the circular type with diffusion vanes. The change from velocity to pressure is accomplished by slowing down the speed of the water, and it is necessary that this change take place gradually and uniformly, with the least loss from eddies and shock. With a proper design of volute or diffusor, it is possible to transform practically the whole of the velocity into pressure so that the loss from this source is very small.

Impellers

Enclosed Impeller

Open Impeller

Fig. 39

There are two general forms of impellers which are known as the open and enclosed type.

The former consists of a set of radial vanes attached to the central hub and disk, and open at the sides, the whole revolving between the two fixed side plates of the pump, while in the latter the vanes are formed between two circular disks which form closed passages between the vanes and extend from the inlet opening to the outer periphery of the impeller. Since the open impeller revolves between the two stationary disks, it is necessary to allow some clearance on each side to prevent contact, with the result that there is considerable leakage at these extended points and a consequent loss of efficiency. The water passing through the impeller is revolved against these stationary side plates with a velocity approximately the same as that of the vanes, and therefore offers a frictional resistance proportional to the square of this velocity. In the enclosed impeller there can be no leakage by the sides of the vanes. The only possible leakage being around the outside of the impeller into the suction, and this is practically prevented from occuring by means of a running fit around the inlet opening. The frictional loss of an enclosed impeller is caused only by the outer surface revolving in the surrounding water. Since the frictional loss of the impeller is the principal loss in the centrifugal pump, it is evident that any saving at this point will greatly improve the efficiency of the pump. For these reasons the enclosed impeller is used more extensively than the open impeller and particularly in efficient pumps. The open impeller is particularly adapted to handling liquids which contain sand, grit, or other foreign matter.

Turbine and Volute Pumps

Centrifugal pumps are divided into two general classes: Turbine and Volute Pumps.

The Turbine Pump

The turbine pump is the type used for heads higher than 250 feet. With this type of pump, 100 feet per stage usually gives satisfactory results, but it is possible to operate on heads up to 300 feet per stage without difficulty, depending upon the speed available.

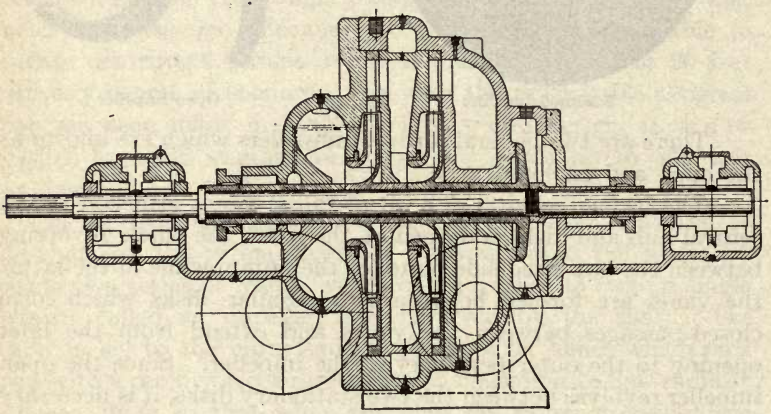


Fig. 40. Multistage Turbine Pump.

The turbine pump has a circular casing, and has diffusion vanes which surround the impeller. These diffusion vanes provide gradually enlarging passages, whose function it is to reduce the velocity of the water leaving the impeller, and efficiently transform the velocity head into pressure head.

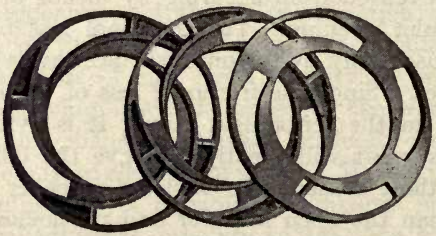


Fig. 42. Diffusion Vanes of Turbine Pump.

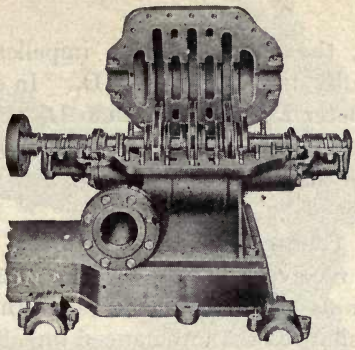


Fig. 189. Multistage Volute Pump.

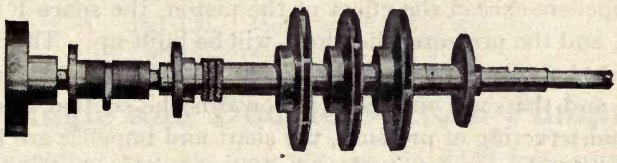


Fig. 43. Shaft and Rotating Parts of Multistage Pump.

With the multistage type of pump, it is necessary to provide a balancing device to take up the end thrust of the impeller.

Referring to the accompanying Fig. 44, we see that in space A and B, the pressure created by the impeller is the same.

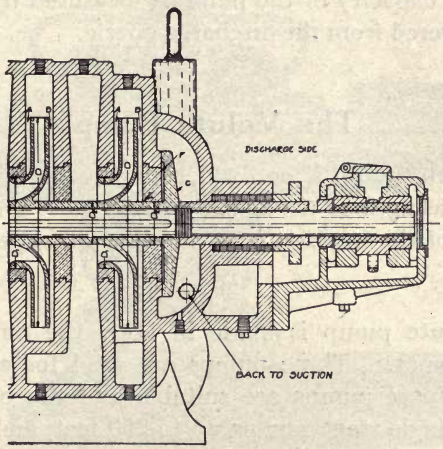


Fig. 44. Hydraulic Balancing Device for Multistage Volute Pump.

In space A, the surface of the impeller exposed to this pressure is shown by diameters D and D_1. In space B, by D and D_2. But as diameter D_2 is smaller than D_1 the thrust on opposite sides will be unequal and the impeller will be pushed towards the suction side of the pump.

In order to counterbalance this effect, there is provided a piston C which receives the pressure along the sleeve E, and is of such diameter as to produce the same thrust in the other direction on the shaft as the unbalanced impellers.

This device works automatically, for should the thrust of the impellers exceed the effect of the piston, the space F will be closed, and the pressure, therefore, will be built up. This pushes the piston away, and produces a leak, whereupon the pressure drops, and the shaft moves back towards the suction side. By this counteracting of pressure, the shaft and impeller are kept in hydraulic balance, avoiding end thrust on the shaft bearings. As there is a continuous leak from the balancing chambers, the piston does not run directly on the metallic surfaces of the casing, but is separated from the latter by a thin film of water.

The amount of water required by this hydraulic balancing device ranges from ½% to 3% of the capacity of the pump; however, the water used is not included in the rating of the pump, as the capacity of the pump is measured by the quantity of water delivered from the discharge nozzle.

The Volute Pump

The volute pump is one which has no diffusion vanes, but instead the casing is of the spiral type, so as to gradually reduce the velocity of the water as it flows from the impeller to the discharge pipe.

The volute pump is made in both the single-stage and multistage types. These pumps are used for all services for which centrifugal pumps are suitable and the operating head varies with single stage pumps up to 250 feet, and in the multistage types up to 1,000 feet.

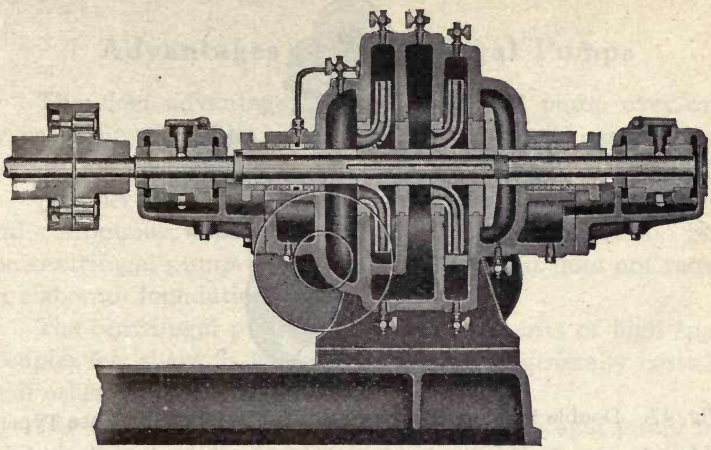


Fig. 190. Multistage Volute Pump with Hydraulic Balancing Device.

Single and Double Suction Pumps

Centrifugal pumps may also be classified into single and double suction according to whether the impeller takes the water from one or both sides. With the single suction pump, there is an unbalanced condition in the impeller, which creates an end thrust, and provision is generally made to take care of this by a thrust bearing of either the hydraulic type or ball thrust type.

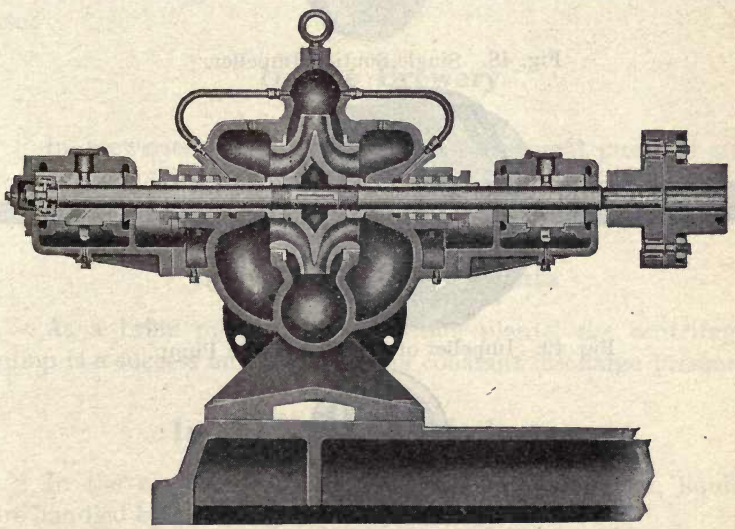


Fig. 191. Double Suction Volute Pump with Ball Thrust Bearing.

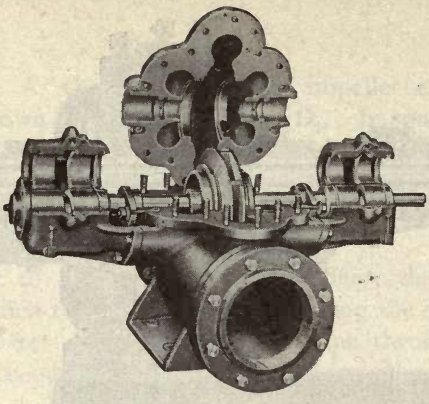


Fig. 47. Double Suction Volute Pump, Horizontal Split Case Type.

In the double suction pump, the impeller takes the water from both sides, and is theoretically balanced, but in practice it has been found that owing to slight variations in castings and local conditions, which exist, this type of impeller is not always balanced, and it is advisable to provide a thrust bearing which is generally of the ball type.

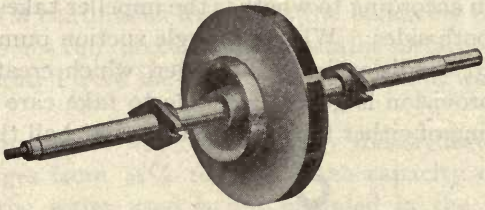


Fig. 48. Single Suction Impeller.

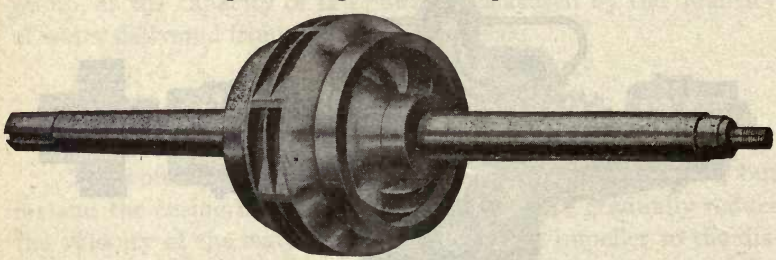


Fig. 49. Impeller of Double Suction Pump.

Fig. 50. Ball Thrust Bearing.

Advantages of Centrifugal Pumps

The chief advantages of the centrifugal pump over other types of pumps are its simplicity, reliability and ease of operation.

Another important feature is that the discharge is smooth and continuous, and free from shocks and pulsations. Since the centrifugal pump is free from vibration, it does not require an elaborate foundation.

The centrifugal pump possessing the merits of high speed, occupies less space, is lighter in weight, and generally costs less than other types of pumps.

The fact that the discharge from a centrifugal pump may be shut off by merely closing a valve in the discharge pipe without dangerous pressures being introduced, or requiring the motor to be shut down, is another great advantage.

Uses of Centrifugal Pumps

The centrifugal pump has reached such a stage of development, that it is being used in almost every conceivable industry in which the use of water and other liquids play a part.

The following brief outline will give an idea of its extensive use:

In the Brewery

(Near-Beer)

In breweries and distilleries, the centrifugal pump is successfully employed in handling hot and cold mash, beer, grain, and for circulating water.

In Refrigerating Plants

As a brine pump in refrigerating plants, the centrifugal pump is a success on account of its constant discharge pressure.

In the Chemical Industry

In the chemical industry, in soap and oil plants, liquids are handled by centrifugals.

For Drainage and Irrigation

Drainage and Irrigation pumps usually involve low heads and generally the amount of power to be supplied is comparatively small. The centrifugal pump for this service is a desirable unit on account of its simplicity, low first cost, and low operating expense.

Elevator Service

For elevator service, which involves a supply of varying quantities of water under a high and practically constant head, the centrifugal pump is particularly adapted.

Fire Pump Service

As a fire pump, the centrifugal is used extensively on account of its exceptional reliability and simplicity.

The power-limiting characteristic is particularly valuable since it protects the driving motor against overloads; also the flat head delivery characteristic will prevent excessive rise in pressure and possible rupture of hose, when the delivery is deminished.

Hot Water Service

For the circulation of hot water in heating systems, the centrifugal pump is extensively used on account of its constant discharge pressure, and because it does not give rise to disturbing noises in the piping system and radiators.

In the House and Office Building

The centrifugal pump is particularly adapted for house and office buildings where noiseless operation is imperative, and constant pressure is desired.

In the Iron and Steel Industry

In the iron and steel industry, centrifugal pumps have become an important factor for handling liquids.

For Marine Work

On board ship for circulating, ballast, general service, condensate and boiler feeding, the centrifugal pump is being used extensively.

For Mine Service

Centrifugal pumps are used in mining work as station pumps, sinking pumps, and for sluicing and hydraulic mining. They are admirably adapted for mine service, as they require little or no foundation, do not produce vibration in the pipe line, involve little cost for installation, and when electrically operated, are easily controlled from a distant point.

In the Oil Industry

Centrifugal pumps are being used extensively in the oil industry for handling oils where the viscosity is such that it is possible. This type of pump is particularly adapted for handling light oils such as light crudes, gas oil, kerosene, gasoline, etc.

Paper Mills

In the paper mill, the centrifugal pump is used extensively for circulating water, handling pulp, etc.

The open impeller pumps are particularly adapted for handling liquids containing solid matter.

In the Power House

In power house work, centrifugal pumps are used for boiler feeding, condensate pumps, circulating pumps with surface, jet and barometric condensers, sump pumps, etc.

For the Sugar House

In sugar houses and refineries, where reliability is the chief requirement, centrifugal pumps are used extensively for water supply, juices, carbonation pumps, filter press pumps, etc.

Water Works Service

For water works service, centrifugal pumps are used for the main pumping units, for booster pumps, and in connection with filter plants, for filling sedimentation basins, flushing filters, etc.

Data Required for Estimates for Union Centrifugal Pumps

When sending for estimates, please answer the following questions:—

1. Number of pumps required.....

2. Capacity of each pump.....U. S. gallons per minute.

3. Total lift, including suction lift, discharge lift, and pipe friction.....feet.

4. Length.....and size.....of suction pipe and maximum distance from water level to pump.....feet.

5. Length.....and size.....of discharge pipe, number.....and type.....of elbows and bends.

6. Nature of liquid to be handled.....Fresh water, salt water, acidulous, alkaline, gritty, solids in suspension?

7. Temperature of liquid.....° Fah. Specific gravity .....

8. Service, continuous..... Intermittent.....

9. If electric-motor-driven, state characteristics of current .....If direct current, give voltage.....If alternating current, give voltage.....cycles.....phase.....

10. If steam driven, state whether connected to steam turbine or steam engine.

11. Give steam pressure, superheat, if any, and state whether condensing or non-condensing.

12. If belt driven, give dimensions and speed of driving pulley.

NOTE.—Give additional information as to location, service of pump, special conditions, etc., in order to enable us to furnish the proper outfit.

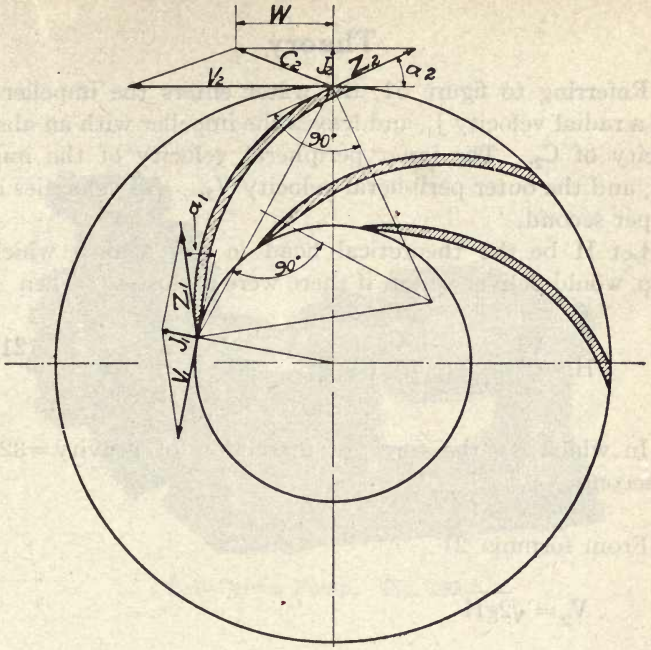


Fig. 51.

Impeller Diagram

In the above diagram

V_2 = Tangential velocity, impeller at outer periphery.

V_1 = Tangential velocity, impeller at inner periphery.

Z_2 = Relative velocity of water at outlet.

Z_1 = Relative velocity of water at inlet.

C_2 = Absolute velocity of water at outlet.

J_2 = Radial velocity of water at outlet.

J_1 = Radial velocity of water at inlet.

W = Tangential velocity of water at outlet.

a_2 = Outlet angle of impeller.

a_1 = Inlet angle of impeller.

The above diagram illustrates the layout of a centrifugal pump impeller. Like all engineering work, the various factors entering into the design of centrifugal impellers are determined by experience. The design of a centrifugal pump impeller is ultimately based on the performances of other impellers. The theory indicates what would be the general effect of altering certain dimensions, hence, successful design consists of modifying or changing the design of impellers, which have been tested out.

Theory

Referring to figure 51, the water enters the impeller inlet with a radial velocity J_1, and leaves the impeller with an absolute velocity of C_2. The inner peripheral velocity of the impeller is V_1, and the outer peripheral velocity V_2. All velocities are in feet per second.

Let H be the theoretical head in feet against which the pump would deliver water, if there were no losses. Then

$$H = \frac{V_2^2}{2g} \qquad (21)$$

In which g = the force of attraction of gravity = 32.2 ft. per second.

From formula 21

$$V_2 = \sqrt{2gH}$$

Having given the head against which the pump must work and the diameter of the impeller, the speed of the pump may be calculated by formula 21.

EXAMPLE: Assume we have to pump against a head of 100 ft., and have an impeller of 10¾" diameter. What would be the required speed of the pump?

By substituting in formula 21, we have:

$$V_2 = \sqrt{2gH}$$

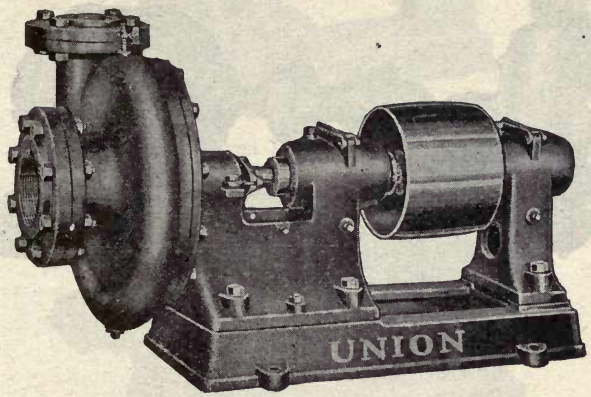
$$V_2 = \sqrt{2 \times 32.2 \times 100} = 80.4 \text{ ft. per second.}$$

This is equal to 80.4 x 60 = 4824 ft. per minute.

The circumference of the impeller 10¾" in diameter = 10¾ x 3.14 = 33.8" or 2.8 ft.

As the impeller has to revolve 4824 ft. per minute, it will have to run $\frac{4824}{2.8}$ = 1722 revolutions per minute.

The capacity of a pump depends upon the size of the suction and discharge openings, the size of the casing, and width and diameter of the impeller. These factors are determined by the designer from experience.

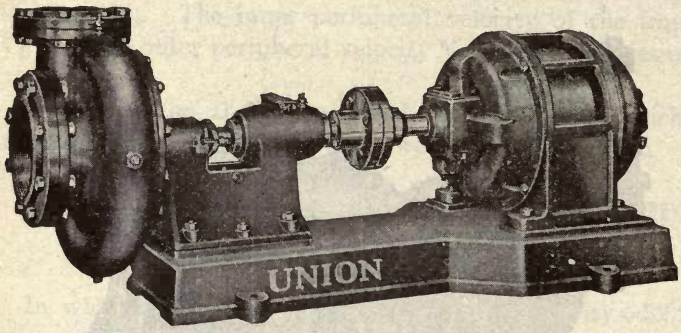


Belt-Driven Pump. Fig. 192.

Union Belt-Driven Side-Suction Volute Pumps

Maximum Working Pressure 52 pounds, or 120 feet.

SIZE OF PUMP	Pipe Sizes		Capacity, G. P. M.			Standard Pulleys	
	Discharge Inches	Suction Inches	Minimum	Normal	Maximum	Diameter, Inches	Face, Inches
¾	¾	1	5	12	16	3	2
1	1	1¼	10	25	35	3	3
1¼	1¼	1½	25	60	90	4	4
1½	1½	2	40	90	125	5	5
2	2	3	75	200	260	6	6
2½	2½	3	125	260	300	6	6
3	3	4	200	400	600	8	6
4	4	5	300	500	700	8	8
5	5	6	400	800	1000	10	8
6	6	8	500	1000	1200	10	10
8	8	10	1200	1500	2000	12	12
10	10	12	1500	3000	3500	14	12
12	12	14	2500	4000	5000	16	14

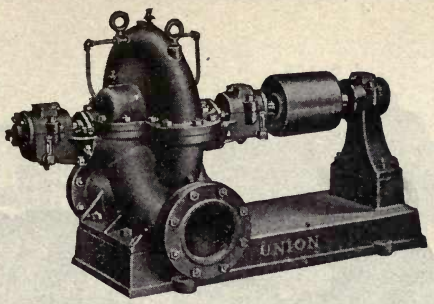


Motor-Driven Pump. Fig. 193.

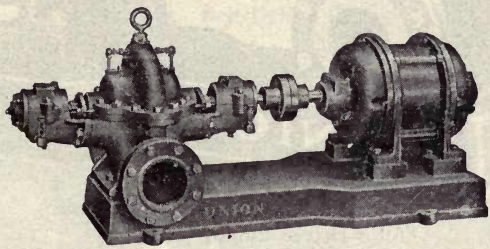
Union Motor-Driven Side-Suction Volute Pumps

Maximum working pressure 52 pounds, or 120 feet.

SIZE OF PUMP	Pipe Sizes		Capacity, G. P. M.		
	Discharge, Inches	Suction, Inches	Minimum	Normal	Maximum
¾	¾	1	5	12	16
1	1	1¼	10	25	35
1¼	1¼	1½	25	60	90
1½	1½	2	40	90	125
2	2	3	75	200	260
2½	2½	3	125	260	300
3	3	4	200	400	600
4	4	5	300	500	700
5	5	6	400	800	1000
6	6	8	500	1000	1200
8	8	10	1200	1500	2000
10	10	12	1500	3000	3500
12	12	14	2500	4000	5000



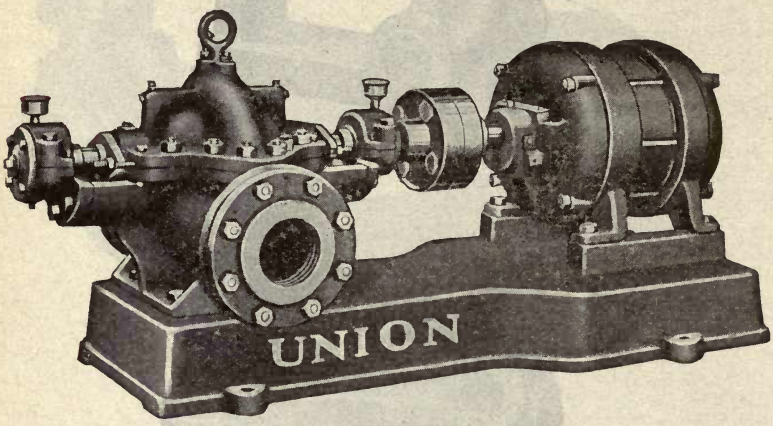
Belt Driven Pump. Fig. 54.



Motor Driven Pump. Fig. 55.

Union Horizontal Double-Suction Volute Pumps
Horizontal Split-Case Type

SIZE OF PUMP	Pipe Sizes			CAPACITY, G. P. M.			Standard Pulleys	
	Discharge, Inches	Suction, Inches	Maximum Head, in Feet	Minimum	Normal	Maximum	Diameter, Inches	Face, Inches
1½ BL	1½	2	100	40	90	125	5	5
2 BS	2	3	100	75	125	150	5	5
2 BL	2	3	150	75	125	150	5	5
3 BS	3	4	100	175	280	325	6	6
3 BL	3	4	200	175	280	325	6	6
4 BS	4	5	100	300	450	600	8	8
4 BL	4	5	200	300	450	600	8	8
5 BS	5	6	100	450	800	950	10	10
5 BL	5	6	200	450	800	950	10	10
6 BS	6	8	100	600	1000	1400	10	10
6 BL	6	8	200	600	1000	1400	10	10
8 BS	8	10	100	1000	2000	2300	12	12
8 BL	8	10	200	1000	2000	2300	12	12
10 BS	10	12	100	1600	3000	4000	14	12
10 BL	10	12	200	1600	3000	4000	14	12
12 BL	12	14	200	2800	4000	5000	15	12
14 BL	14	16	200	3500	5500	6500	16	14



Motor-Driven Pump. Fig. 194.

Union Double-Suction Volute Pumps

High Speed.

For Maximum Heads of 250 feet at 3500 R. P. M., also for Low Heads at 1750 R. P. M.

SIZE OF PUMP	PIPE SIZES		CAPACITY, G. P. M.		
	Discharge, Inches	Suction, Inches	Minimum	Normal	Maximum
2	2	3	100	200	300
3	3	4	200	400	500
4	4	5	300	500	750
5	5	6	500	800	1000
6	6	8	800	1200	1400

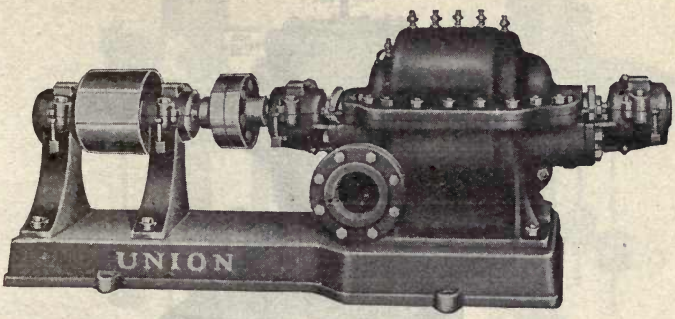


Fig. 195. Belt-Driven Pump.

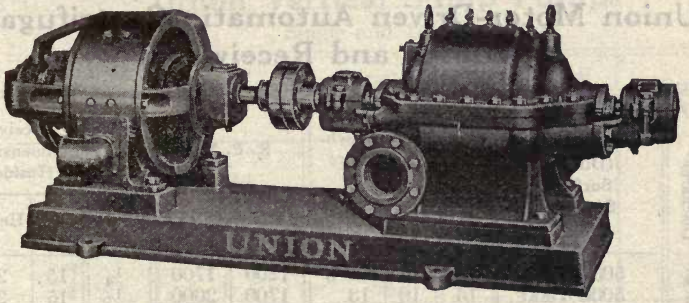


Fig. 196. Motor-Driven Pump.

Union Multistage Centrifugal Pumps

SIZE OF PUMP	PIPE SIZES		CAPACITY, G. P. M.	
	Discharge, Inches	Suction, Inches	Minimum	Maximum
1½	1½	2	40	120
*2HS	2	3	50	100
2	2	3	50	175
3	3	4	150	350
4	4	5	300	650
5	5	6	450	900
6	6	6	600	1200

*This size only has ball-bearings.

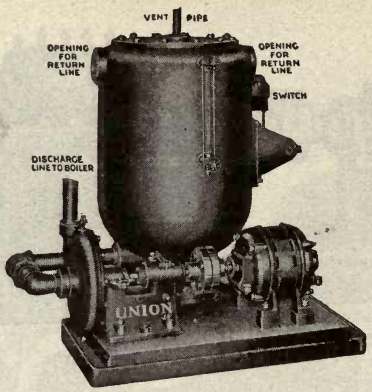


Fig. 179.

Union Motor-Driven Automatic Centrifugal Pumps and Receivers

Size Pump Discharge	**Radiation (Direct) Surface Sq. Feet	G. P. M.	Max. Disch. ††Pressure		R. P. M.*		H. P.† MOTOR	Receiver Dimensions (Inside)	
			Lbs.	Feet Head	Min.	Max.		Dia.	Height
¾	5000–7500	15	5	11.5	1200	1700	¼	15	21
¾	5000–7500	15	10	13	1700	2000	½	15	21
¾	5000–7500	15	15	34.5	2000	2500	¾	15	21
1	7500–12000	24	5	11.5	1100	1700	½	15	21
1	7500–12000	24	10	23	1300	1700	¾	15	21
1	7500–12000	24	15	34.5	1600	2000	1	15	21
1	7500–12000	24	20	46	1750	2500	1	15	21
1¼	12000–30000	60	5	11.5	1100	1700	¾	22	36
1¼	12000–30000	60	10	23	1150	1700	1	22	36
1¼	12000–30000	60	15	34.5	1350	1700	1½	22	36
1¼	12000–30000	60	20	46	1500	1700	3	22	36
1¼	12000–30000	60	25	57.5	1700	2000	3	22	36
1¼	12000–30000	60	30	69	1800	2000	3	22	36
1½	30000–40000	80	5	11.5	1100	1400	1½	22	36
1½	30000–40000	80	10	23	1100	1700	1½	22	36
1½	30000–40000	80	15	34.5	1100	1700	2	22	36
1½	30000–40000	80	20	46	1100	1700	3	22	36
1½	30000–40000	80	25	57.5	1300	1700	5	22	36
1½	30000–40000	80	30	69	1400	1700	5	22	36
1½	30000–40000	80	35	81	1500	1700	5	22	36

*RPM refers to the full-load speed of the motor.
**For indirect radiation as in fan system, one-fifth of the ratings given should be used.
†Use motors of 40° Centrigrade rating.
††In figuring total head, allow 5 lbs. margin for forcing water into the boiler.
For higher pressure, special pumps can be furnished upon request.

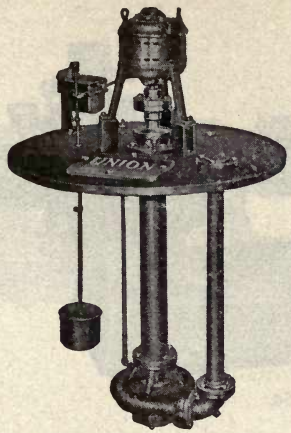


Fig. 180. Single Sump Pump.

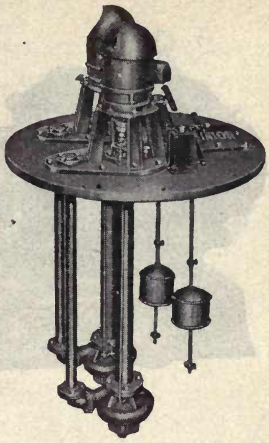


Fig. 197. Duplex Sump Pump.

Union Vertical Sump Pumps

Single and Duplex.

SIZE PUMP	Size Discharge	Diameter of Cover in Inches Single	Diameter of Cover in Inches Duplex	‡Depth of Pit in Feet
**1	1	30	48	6
1¼	1¼	36	60	6
1½	1½	42	60	6
2	2	42	60	6
2½	2½	48	68	6
3	3	48	68	6
4	4	52	68	6

**This pump is not suitable for handling soil.

‡For pits deeper than 6 feet, steady bearings are used.

Capacity, Speed and Horse Power Table for Vertical Sump Pumps

HEAD IN FEET

G. P. M	10			20			30			40			50		
	Size Pump	R. P. M.	H. P. Motor	Size Pump	R. P. M.	H. P. Motor	Size Pump	R. P. M.	H. P. Motor	Size Pump	R. P. M.	H. P. Motor	Size Pump	R. P. M.	H. P. Motor
10	1	1150	¼	1	1740	½	1	1740	¾	1	1740	1	1¼	1740	1½
20	1	1150	¼	1	1740	¾	1	1740	¾	1	1740	1	1¼	1750	2
30	1	1150	½	1	1740	¾	1	1740	1	1	1740	1½	1¼	1750	2
40	1¼	1150	½	1¼	1150	1	1¼	1740	1½	1¼	1740	2	1¼	1750	2
50	1¼	1550	¾	1¼	1750	1	1¼	1740	1½	1¼	1750	2	1¼	1750	3
75	1½	1150	1	1½	1150	1½	1¼	1750	2	1¼	1750	3	1¼	1750	3
100	1½	1150	1½	1½	1150	1½	1½	1150	3	1½	1750	5	1½	1750	5
150	2	1150	1½	2	1150	2	2	1150	3	2	1150	5	2	1750	5
200	2	1150	1½	2	1150	3	2	1150	3	2	1150	5	2	1750	7½
250	2½	1150	2	2½	1150	3	2½	1150	5	2½	1150	5	2	1750	7½
300	3	1150	2	3	1150	5	½	1150	5	2½	1150	7½	3	1150	7½
350	3	1150	2	3	1150	5	3	1150	5	3	1150	7½	3	1150	10
400	3	1150	3	3	1150	5	3	1150	7½	3	1150	7½	4	1150	10
500	4	860	3	4	1150	5	4	1150	7½	4	1150	10	4	1150	15
600	4	860	5	4	1150	7½	4	1150	10	4	1150	10	4	1150	15

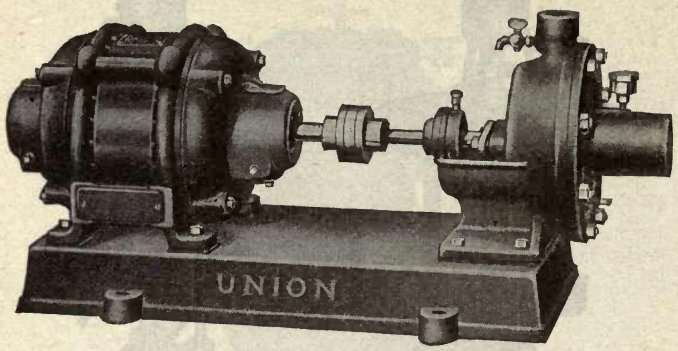


Fig. 186.

Union Two-Stage House Pump

G. P. M.		10	15	20	25	30	35	40	45	50
Speed, Max. Min. R. P. M.		Pounds Pressure or Foot Elevation and H. P. of Motor Required								
Max. 3600 Min. 3400	Pressure Elevation					78 180	71.5 165	65 150	61.5 142	50 115
	Motor H.P.					5	5	5	5	5
Max. 3600 Min. 3200	Pressure Elevation				72.5 167	68 157	60.5 140	54 125	47.5 110	
	Motor H.P.				3	3	3	3	3	
Max. 3600 Min. 3000	Pressure Elevation				63 145	58.5 135	52 120	45.5 105	39 90	
	Motor H.P.				3	3	3	3	3	
Max. 3400 Min. 2800	Pressure Elevation			56 130	52.5 121	47.7 110	42 97	36 83		
	Motor H.P.			2	2	2	2	2		
Max. 3400 Min. 2600	Pressure Elevation			47.7 110	43.3 100	39 90	32.5 75	27 62		
	Motor H.P.			2	2	2	2	2		
Max. 3000 Min. 2400	Pressure Elevation		41 95	39 90	34.7 80	30.3 70	26 60			
	Motor H.P.		1½	1½	1½	1½	1½			
Max. 2800 Min. 2200	Pressure Elevation		34.6 80	32.5 75	27 62	24 55				
	Motor H.P.		1	1	1	1	1½			
Max. 2500 Min. 2000	Pressure Elevation		27 63	26 60	21.6 50					
	Motor H.P.		1	1	1					
Max. 2500 Min. 1700	Pressure Elevation	21.6 50	19.5 45	16.5 38						
	Motor H.P.	¾	¾	¾						

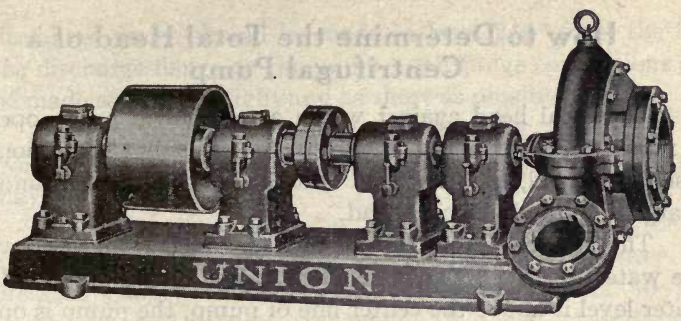


Fig. 198.
Belt-Driven Pump.

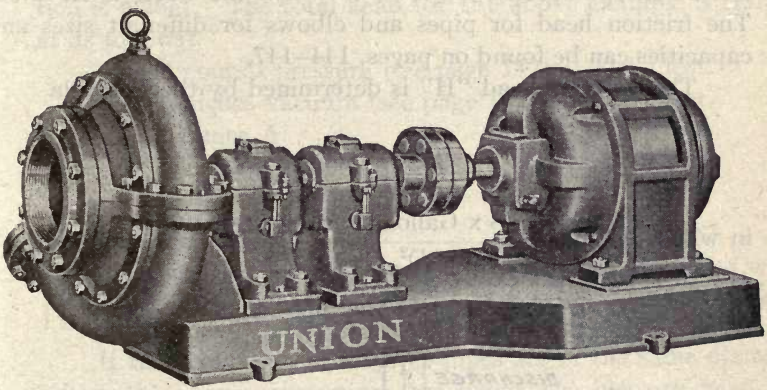


Fig. 199.
Motor-Driven Pump.

Union Centrifugal Paper Stock Pumps

SIZE OF PUMP	PIPE SIZES		*APPROXIMATE CAPACITY, G. P. M.		Standard Pulleys	
	Discharge, Inches	Suction, Inches	Minimum	Maximum	Diameter, Inches	Face, Inches
3	3	5	150	350	8	6
4	4	6	250	700	10	8
5	5	8	400	900	12	8
6	6	10	600	1200	12	10
8	8	12	900	2000	14	12

*Capacity will vary with the consistency of stock.

How to Determine the Total Head of a Centrifugal Pump

The total head against which a centrifugal pump operates is made up of the sum of four factors as follows: suction lift, discharge head, friction head (due to loss in suction and discharge line), and velocity head.

The suction lift is the vertical distance from the level of the water to be pumped to the center line of the pump. If the water level is above the center line of pump, the pump is operating under a suction head or a flooded suction, and this distance must be subtracted from the sum of the remaining factors. The discharge head is the vertical distance between the center line of the pump and the level to which the water is elevated. The friction head for pipes and elbows for different sizes and capacities can be found on pages, 144–147.

The velocity head "H" is determined by the Formula

$$H = \frac{V_2}{64.4} \qquad (22)$$

in which $V = \dfrac{.408 \times \text{Gallons per minute}}{D^2}$

D = Diameter of the pipe in inches.

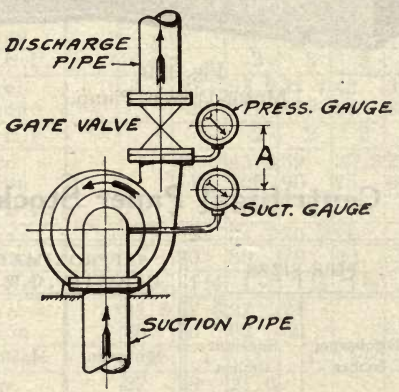


Fig. 61.

Figure 61, illustrates the proper method of connecting up a centrifugal pump for testing purposes.

Connection for suction gauge should be made at least 2½" from the face of the suction flange on pump. Connection for

discharge gauge should be made at least 2½" from the face of the discharge flange on the pump. All gauge connections should be made absolutely tight and as short as possible.

To arrive at the total head that the centrifugal pump works against, from the gauge readings, the following example may be used:

Assuming the distance "A" (vertical distance from the center line of the gauge connection in suction pipe to center line of pressure gauge) to be 2 feet, discharge pressure 40 pounds (by gauge), and vacuum (by gauge) 15 inches, when discharging 1,000 gallons of water per minute. Let 6 inches be the diameter of the discharge pipe (where gauge connection is made) and 8 inches be the diameter of the suction pipe (where gauge connection is made). The total head for the above example is arrived at as follows:

40 pounds pressure (see page 142)	=92.4 feet.
15 inches vacuum (see page 152)	=17.01 feet.
Distance A	= 2.0
*Velocity head	= 1.36.
Total head	=112.77 feet.

*The velocity head in the 6 inch discharge pipe by formula (22) equals 1.99 feet. The velocity head in the 8 inch suction pipe by formula (22) equals .63 feet. The total velocity head to be added, therefore equals the difference between these two figures or 1.36 feet.

If the suction and discharge pipes are of the same diameter where the gauge connections are made, the velocity head will be the same in both, and no correction need be made for same, as the suction gauge readings include the velocity head in the suction pipe, which in this instance is the same as the velocity head on the discharge pipe. Where the discharge pipe is smaller in diameter than the suction pipe, the difference between the velocity heads in both pipes should be added to the other readings given above in order to arrive at the total head. The difference in velocity heads in the suction and discharge pipes should be subtracted from the sum of the other readings given above, if the suction pipe is smaller than the disharge pipe where the gauges are connected. In the above example, the friction head in the suction and discharge pipes is included in the gauge readings.

It is of the greatest importance that the correct head be determined, before purchasing a centrifugal pump.

By referring to the characteristics reproduced on page 117, it is seen that should the head be greater than that for which the pump is designed, less water would be discharged, and if the head is less, more water will be discharged. Particularly on direct connected units, any error in specifying the correct head involves either changing the impeller, the prime mover, or both.

Great care should be used in estimating the friction loss in the piping, as this may be a very important factor of the total head.

Measurement of Water

To determine the volume of discharge of a centrifugal pump or any pump, the means that may be employed according to the circumstances are to weigh or measure the volume of liquid discharged in a known time interval, by using the weir, a Venturi meter, a Pitot tube, or a calibrated nozzle.

To measure the volume of, or weigh the liquid discharged in a certain time interval, is the most accurate method, owing to the fact that no arbitrary constants are necessary in calculation.

This method of measurement is very often used in laboratories and is the one used in testing out Union centrifugal pumps.

The Union Steam Pump test laboratory is equipped with a large testing tank containing approximately 60,000 gallons of water. The pump takes the suction from this tank and discharges it into a smaller tank of exact known dimensions.

The weir is a standard device for measuring water. It should be remembered, however, that all weir formulas and coefficients are purely empirical in their nature, and that the different formulas that are accepted at large do not give identical results. The most widely used weir formula is the Francis formula for rectangular weirs.

$$Q = 3.33\ (b - .2H)\ H^{\frac{3}{2}} \qquad (23)$$

Q = Cubic feet per second.

b = Breadth in feet of the notch or length of the weir.

H = The head in feet above the crest measured by the hook gauge.

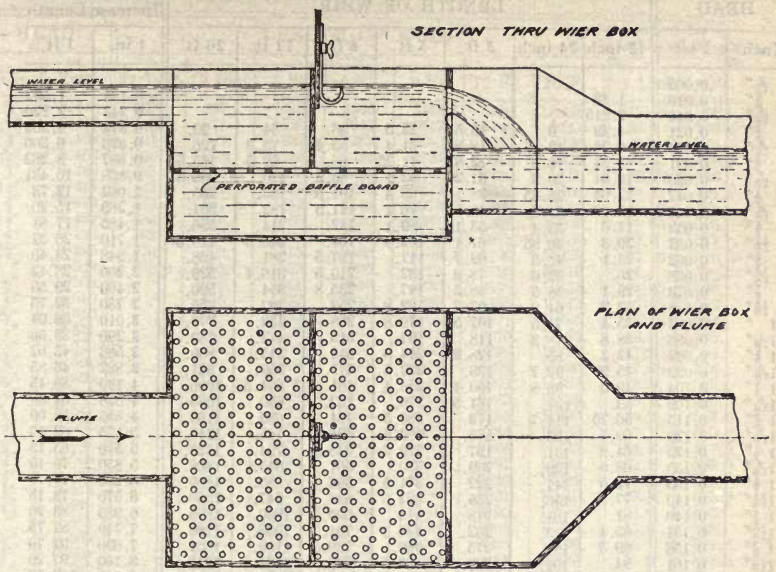


Fig. 62.

Standard Full Contracted Rectangular Weir Box for Measurement of Water

Figure 62 illustrates a standard full contracted weir box for the measurement of water.

In constructing the same, it is necessary that the board over which the water falls, be beveled on the down stream side; the ends should also be beveled on the same side, leaving the edge almost sharp, say within one-eighth of an inch. The hook gauge must first be set at zero when the point of hook is level with the crest of weir by use of a spirit level. Or it can be set perfectly accurate with the water level just at the crest of the weir and point of hook showing above the water.

The tables on the following pages are based on zero velocity of approach, i. e., the water should not approach the weir with any noticeable velocity, as otherwise a greater quantity would be discharged than indicated by the depth.

The length of the weir should be less than two-thirds the width of the box, and the depth of the box should be more than three times the depth of water flowing over the crest of the weir.

Francis says a fall below the crest of the weir of one-half the head is sufficient, but there must be a free access of air under the sheet. However, we recommend that the fall below the weir should be greater than one-half the head.

Discharge of Rectangular Weir

HEAD		LENGTH OF WEIR							Addition for Increase Length	
Inch	Feet	12 inch	24 inch	3 ft.	5 ft.	8 ft.	12 ft.	20 ft.	1 in.	1 ft.
1/16"	0.005									
1/8"	0.010	1.48							0.138	1.592
3/16"	0.015	2.915							0.243	2.220
1/4"	0.021	4.49	9.	13.5	22.5	36.	54.	90.	0.374	4.505
5/16"	0.026	6.25	12.55	18.8	31.4	50.4	75.6	126.	0.526	6.300
3/8"	0.031	8.24	16.5	22.6	41.25	66.	99.5	165.5	0.687	8.282
7/16"	0.036	10.34	20.8	31.2	52.1	83.2	125.	208.3	0.868	10.421
1/2"	0.042	12.64	25.35	38.1	63.4	101.8	153.	255.	1.062	12.73
9/16"	0.047	15.08	30.3	45.5	76.1	121.5	182.	304.	1.269	15.20
5/8"	0.052	17.6	35.4	53.1	89.1	142.	214.	356.	1.485	17.81
11/16"	0.057	20.3	40.85	61.1	102.8	164.	246.5	410.5	1.710	20.57
3/4"	0.062	23.1	46.3	69.7	117.	187.5	281.	468.	1.948	23.40
13/16"	0.068	26.	52.6	78.9	132.	210.5	316.4	529.	2.200	26.42
7/8"	0.073	29.1	58.5	88.3	147.	235.8	354.	590.	2.460	29.50
15/16"	0.078	32.2	64.8	97.7	162.8	252.	392.	655.	2.730	32.75
1 "	0.083	35.4	71.5	107.5	179.8	288.	432.	770.	3.010	36.04
1 1/16"	0.088	38.8	78.3	118.	197.	315.	472.	789.	3.290	39.50
1 1/8"	0.094	42.2	85.	128.2	214.5	343.	515.	860.	3.583	43.02
1 3/16"	0.099	45.9	92.2	139.	232.5	372.	559.	931.	3.882	46.85
1 1/4"	0.104	49.5	99.8	150.4	250.4	401.	604.5	1006.	4.199	50.45
1 5/16"	0.109	53.	107.	161.5	270.	432.	650.	1085.	4.520	54.00
1 3/8"	0.115	56.75	114.7	173.	289.5	464.	695.	1160.	4.855	58.00
1 7/16"	0.120	60.7	123.	185.	309.5	496.	745.	1240.	5.175	62.10
1 1/2"	0.125	64.9	131.	197.	329.5	528.	794.	1323.	5.570	65.15
1 9/16"	0.130	68.5	139.	209.	350.	561.	845.	1408.	5.850	70.10
1 5/8"	0.135	72.5	147.	222.	371.5	596.	885.	1500.	6.203	74.70
1 11/16"	0.140	77.	156.	235.	392.6	630.	947.	1580.	6.570	79.15
1 3/4"	0.146	81.	164.	248.	415.	665.	1000.	1680.	6.985	83.20
1 13/16"	0.151	85.4	173.	262.	436.5	701.	1053.	1760.	7.340	87.75
1 7/8"	0.156	89.5	182.	275.	460.	736.	1109.	1852.	7.690	92.70
1 15/16"	0.161	94.	191.	289.	483.5	775.	1164.	1942.	8.100	97.20
2 "	0.167	98.5	200.5	302.	506.	812.	1220.	2040.	8.515	102.00
2 1/16"	0.172	103.	210.	316.	530.	850.	1278.	2130.	8.911	106.80
2 1/8"	0.177	107.8	219.9	332.	555.	890.	1330.	2230.	9.316	111.70
2 3/16"	0.182	112.4	229.	345.	579.	930.	1397.	2330.	9.715	116.60
2 1/4"	0.187	117.	239.	361.	605.	970.	1453.	2430.	10.122	121.30
2 5/16"	0.193	122.	249.	376.	629.	1010.	1518.	2530.	10.590	127.00
2 3/8"	0.198	127.	259.	390.5	655.	1050.	1580.	2637.	10.990	132.00
2 7/16"	0.203	132.	269.	406.	680.	1092.	1640.	2738.	11.420	137.20
2 1/2"	0.208	136.2	279.	422.	706.	1133.	1707	2846.	11.890	142.90
2 9/16"	0.213	142.	289.	438.	732.	1176.	1769.	2955.	12.320	148.00
2 5/8"	0.219	146.7	300.	453.	760.	1220.	1832.	3057.	12.790	153.50
2 11/16"	0.224	151.4	310.5	470.	785.	1265.	1900.	3165.	13.210	159.00
2 3/4"	0.229	157.	321.5	485.	815.	1308.	1968.	3275.	13.590	164.20
2 13/16"	0.234	162.	332.	501.5	832.5	1352.	2034.	3470.	14.19	170.15
2 7/8"	0.240	167.6	343.	520.	870.	1400.	2103.	3500.	14.61	175.3
2 15/16"	0.245	172.9	354.	535.	898.	1442.	2171.	3611.	15.05	181.4
3 "	0.250	177.8	366.	552.	926.	1490.	2239.	3740.	15.61	187.3
3 1/16"	0.255	183.3	377.	569.	956.	1536.	2309.	3853.	16.1	193.5
3 1/8"	0.260	189.1	388.	588.	986.	1580.	2380.	3975.	16.6	199.2
3 3/16"	0.266	194.8	400.	605.	1015.	1632.	2449.	4092.	17.1	205.5
3 1/4"	0.271	199.8	410.5	624.	1047.	1679.	2522.	4210.	17.6	211.
3 5/16"	0.276	205.6	422.	640.	1076.	1728.	2598.	4341.	18.11	217 6
3 3/8"	0.281	210.8	435.	659.	1105.	1778.	2671.	4455.	18.63	224.
3 7/16"	0.286	216.5	446.	676.	1138.	1825.	2740.	4575.	19.11	229.8
3 1/2"	0.292	222.	458.	695.	1167.	1875.	2820.	4710.	19.7	236.15
3 9/16"	0.297	228.	470.	714.	1200.	1925.	2898.	4846.	20.2	242.2
3 5/8"	0.302	234.	483.	731.	1230.	1977.	2970.	4961.	20.78	249.4
3 11/16"	0.307	240.	445.	750.	1260.	2027.	3043.	5100.	21.3	255.9
3 3/4"	0.312	245.	506.	769.	1292.	2081.	3121.	5213.	21.83	262.2
3 13/16"	0.318	251.	520.	789.	1328.	2128.	3203.	5350.	22.4	268.6
3 7/8"	0.323	256.5	533.	808.	1355.	2180.	3280.	5475.	22.9	274.5
3 15/16"	0.328	263.	545.	825.	1390.	2239.	3360.	5610.	32.5	282.
4 "	0.333	269.	556.	846.	1424.	2288.	3440.	5748.	24.	288.
4 1/16"	0.338	275.6	570.	866.	1454.	2342.	3520.	5890.	24.64	296.
4 1/8"	0.344	281.6	584.	885.	1490.	2399.	3633.	6015.	25.18	301.9
4 3/16"	0.349	286.	596.	906.	1523.	2450.	3680.	6150.	25.78	309.
4 1/4"	0.354	293.6	610.	925.	1559.	2505.	3775.	6300.	26.34	316.
4 5/16"	0.359	300.	623.	945.	1590.	2560.	3856.	6425.	26.94	323.
4 3/8"	0.365	306.	636.	966.	1628.	2620.	3935.	6571.	27.55	330.5
4 7/16"	0.370	312.	650.	986.	1660.	2670.	4015.	6715.	28.05	336.8
4 1/2"	0.375	318.	663.	1006.	1696.	2704.	4102.	6857.	28.65	344.
4 9/16"	0.380	325.	676.	1030.	1730.	2780.	4195.	7002.	29.25	351.5
4 5/8"	0.385	331.	690.	1050.	1768.	2841.	4275.	7150.	29.85.	358.
4 11/16"	0.390	336.6	704.	1069.	1801.	2899.	4355.	7291.	30.46	366.
4 3/4"	0.396	344.	717.5	1091.	1835.	2958.	4450.	7448.	31.15	374.
4 13/16"	0.401	350.	731.	1111.	1875.	3010.	4540.	7588.	31.75	390.9
4 7/8"	0.406	356.6	744.5	1131.	1908.	3075.	4621.	7710.	32.36	388.2
4 15/16"	0.411	363.7	759.	1156.	1948.	3132.	4710.	7891.	33.	395.9
5 "	0.417	370.	772.	1175.	1985.	3192.	4810.	8039.	33.65	404.3
5 1/16"	0.422	376.5	785.	1200.	2018.	3256.	4896.	8172.	34.2	410.5
5 1/8"	0.427	382.5	800.	1220.	2030.	3313.	4999.	8345.	34.93	419.4
5 3/16"	0.432	388.	815.	1239.	2094.	3368.	5070.	8461.	35.5	426.
5 1/4"	0.437	395.5	830.	1262.	2130.	3439.	5165.	8643.	36.17	434.4
5 5/16"	0.443	401.	844.	1285.	2168.	3500.	5255.	8800.	36.79	441.

Discharge of Rectangular Weir

HEAD		LENGTH OF WEIR							Addition for Increase of Length	
Inch	Feet	12 Inch	24 Inch	3 Ft.	5 Ft.	8 Ft.	12 Ft.	20 Ft.	1 Inch	1 Ft.
5 3/8"	0.448	409.	857.	1310.	2208.	3548.	5350.	8949.	37.45	450.
5 7/16"	0.453	415.	871.	1330.	2243.	3612.	5446.	9111.	38.1	457 5
5 1/2"	0.458	421.6	887.	1352.	2282.	3680.	5550.	9250.	38.8	465 5
5 9/16"	0.463	428.5	903.	1376.	2321.	3724.	5612.	9377.	39.2	470.9
5 5/8"	0.469	435.5	915.	1395.	2358.	3780.	5710.	9550.	39.98	480.
5 11/16"	0.474	442.5	932.5	1419.	2400.	3842.	5802.	9706.	40.61	487.6
5 3/4"	0.479	449.	947.5	1442.	2440.	3913.	5899.	9846	41.22	495.
5 13/16"	0.484	456.2	960.	1465.	2480.	3971.	5990.	10000.	41 95	503.5
5 7/8"	0.490	462.6	977.	1490.	2514.	4043.	6078.	10186.	42.6	511.
5 15/16"	0.495	470.	993.	1515.	2559.	4105.	6176.	10325.	43.26	519.6
6 "	0.500	476.5	1005.	1535.	2600.	4161.	6289.	10510.	44.	528.
6 1/16"	0.505		1021.	1561.	2640.	4230.	6398.	10657.	44.65	536.
6 1/8"	0.510		1039.	1582.	2675.	4292.	6490.	10812.	45.25	543
6 3/16"	0.515		1051.	1609.	2716.	4361.	6571.	10996.	46.10	554.
6 1/4"	0.521		1068.	1632.	2760.	4423.	6670.	11175.	46.6	560.
6 5/16"	0.526		1083.	1655.	2801.	4500.	6796.	11321.	47.5	570.
6 3/8"	0.531		1100.	1679.	2844.	4558.	6890.	11500.	47.95	575.
6 7/16"	0.536		1112.	1704.	2881.	4641.	6972.	11680.	48.81	586.
6 1/2"	0.542		1130.	1742.	2920.	4710.	7062.	11825.	49.7	596.
6 9/16"	0.547		1147.	1752.	2962.	4762.	7189.	11990.	50.2	602.3
6 5/8"	0.552		1161.	1779.	3005.	4830.	7288.	12162.	51.1	613.8
6 11/16"	0.557		1178.	1803.	3047.	4900.	7376.	12325.	51.5	617.8
6 3/4"	0.563		1193.	1826.	3094.	4957.	7487.	12500.	52.5	629.9
6 13/16"	0.568		1210.	1853.	3139.	5041.	7590.	12695.	53.3	640.
6 7/8"	0.573		1226.	1878.	3180.	5100.	7691.	12850.	53.8	645.5
6 15/16"	0.578		1240.	1903.	3219.	5181.	7790.	13141	54.6	656.
7 "	0.583		1258.	1928.	3260.	5230.	7902.	13220.	55.6	668.
7 1/16"	0.589		1272.	1949.	3300.	5310.	8005.	13400.	56.	672.5
7 1/8"	0.594		1290.	1976.	3342.	5395.	8112.	13580.	56.9	682.5
7 3/16"	0.599		1309.	2000.	3384.	5450.	8221.	13755.	57.85	695.
7 1/4"	0.604		1322.	2029.	3436.	5561.	8310.	13931.	58.47	701.5
7 5/16"	0.609		1339.	2058.	3480.	5595.	8441.	14110.	59.10	710.
7 3/8"	0.615		1356.	2080.	3522.	5661.	8540.	14290.	59.93	719.6
7 7/16"	0.620		1371.	2105.	3576.	5742.	8641.	14480.	60.6	729.
7 1/2"	0.625		1345.	2130.	3609.	5807.	8750.	14640.	61.4	736.
7 9/16"	0.630		1408.	2155.	3658.	5885.	8850.	14820.	62.1	746.
7 5/8"	0.635		1423.	2179.	3700.	5950.	8980.	15015.	62.96	755.
7 11/16"	0.641		1439.	2212.	3745.	6019.	9093.	15190.	63.6	764.
7 3/4"	0.646		1458.	2238.	3785.	6100.	9185.	15395.	64.49	774.
7 13/16"	0.651		1471.	2260.	3820.	6168.	9291.	15583.	65.15	782.
7 7/8"	0.656		1490.	2286.	3860.	6248.	9419.	15751.	66.	792.
7 15/16"	0.661		1506.	2310.	3903.	6310.	9537.	15921.	66.9	803.
8 "	0.667		1522.	2338.	3956.	6400.	9644.	16150.	67.7	813.5
8 1/16"	0.672		1541.	2365.	4000.	6481.	9746.	16342.	68.5	821.5
8 1/8"	0.667		1555.	2396.	4045.	6540.	9866.	16510.	69.15	831.
8 3/16"	0.682		1572.	2419.	4090.	6609.	9970.	16700.	70.	840.
8 1/4"	0.688		1592.	2442.	4140.	3700.	10095.	16900.	70.8	850.
8 5/16"	0.693		1601.	2460.	4178.	6757.	10195.	17086.	71.6	860.
8 3/8"	0.698		1618.	2493.	4227.	6845.	10310.	17271.	72.55	871.
8 7/16"	0.703		1636.	2516	4292.	6925.	10425.	17905.	73.25	879.6
8 1/2"	0.708		1652.	2540.	4312.	6995.	10530.	17685.	74.03	889.7
8 9/16"	0.174		1670.	2565.	4362.	7061.	10645.	17845.	74.9	899.
8 5/8"	0.719		1689.	2595.	4415.	7146.	10780.	18042.	75.65	909.
8 11/16"	0.724		1706.	2623.	4460.	7225.	10900.	18220.	76.52	919.
8 3/4"	0.729		1723.	2656.	4511.	7293.	11000.	18460.	77.35	929.
8 13/16"	0.734		1741.	2680.	4552.	7380.	11120.	18651.	78.28	940.
8 7/8"	0.740		1760.	2705.	4600.	7460.	11250.	18872	79.	949.
8 15/16"	0.745		1777.	2739.	4648.	7545.	11355.	19000.	79.9	959.
9 "	0.750		1791.	2765.	4699.	7600.	11500.	19206.	80.65	969.5
9 1/16"	0.755		1810.	2792.	4749.	7682.	11600.	19400.	81.52	979.
9 1/8"	0.760		1830.	2816.	4799.	7755.	11720.	19620.	82.9	995.
9 3/16"	0 765		1848.	2844.	4849.	7850.	11831.	19819.	83.11	999.
9 1/4"	0.771		1866.	2876.	4899.	7910.	11940.	20031.	84.25	1011.
9 5/16"	0.776		1880.	2900	4949.	8000	12070.	20225.	85.05	1020.
9 3/8"	0.781		1898.	2927.	4999.	8080.	12190.	20425.	85.92	1031.
9 7/16"	0.786		1918.	2960.	5049.	8160.	12300	20625.	86.76	1041.
9 1/2"	0.792		1939.	2985.	5098.	8241.	12430.	20865.	87.6	1051.
9 9/16"	0.797		1955.	3017.	5145.	8305.	12570.	21035.	88.3	1060.
9 5/8"	0.802		1971.	3041.	5185.	8396.	12695.	21221.	89.45	1072.
9 11/16"	0.807		1989.	3073.	5227.	8483.	12800.	21418.	90.25	1082.
9 3/4"	0.812		2006.	3101.	5288.	8564.	12945.	21625.	91.	1091.
9 13/16"	0.818		2025.	3129.	5340.	8635.	13050.	21825.	91.95	1103.
9 7/8"	0.823		2045.	3160.	5393.	8710.	13190.	22083.	92.85	1112.
9 15/16"	0.828		2065.	3190.	5443.	8800.	13300.	22275.	93.80	1125.
10 "	0.833		2085.	3216.	5490.	8892.	13430.	22532.	94.6	1136.

EXAMPLE—Suppose weir length is 5 ft., and after the water has been flowing sometime, the head or depth of flow is found to be 5 3-16 inches. To find the number of gallons flowing per minute thru the weir, run down the "Head" column until 5 3-16 inches is reached, then crosswise to the column labeled 5 ft., and the flow is found to be 2094 gallons per minute.

If the weir has a different length than any given in the table, such as 73 inches, or 5 ft+1 ft.+1 inch, proceed as follows:—Taking the "Head" the same as in the preceding example, we find the flow for 5 ft. equals 2094, and continuing across on the same line to the columns labeled "Addition for increase of length," we find the flow for a 73 inch weir for the given "Head" is 2094 plus 416 plus 35.5 equals 2555.5 gallons per minute.

For small rates of discharge, the triangular weir is better than the rectangular weir. Any angle of notch may be employed, but the 90° triangular notch is the one most used. The formula for the 90° notch is:

$$Q = 2.544\ H^{\frac{5}{2}} \qquad (24)$$

Q = Cubic feet per second.

H = Head in feet.

The following table has been computed from the above formula:

Head in ft.	G. P. M. discharge	Head in ft.	G. P. M. discharge
0.25	36	0.65	389
0.30	56.2	0.70	468
0.35	83	0.75	555
0.40	115	0.80	655
0.45	155	0.85	760
0.50	202	0.90	879
0.55	256	0.95	1000
0.60	318	1.00	1140

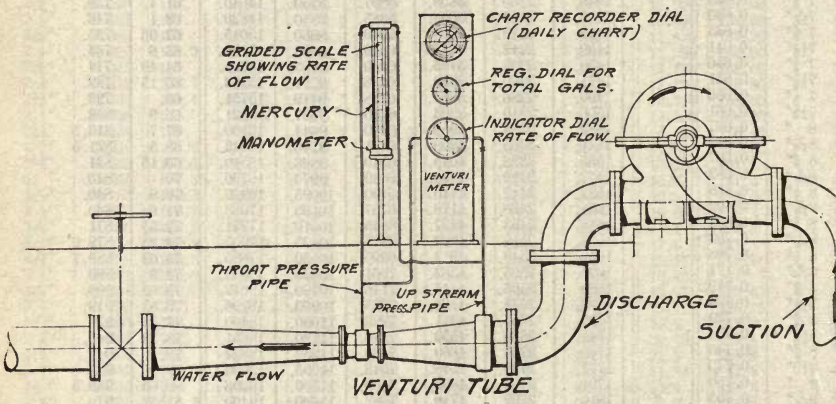


Fig. 63.

The Venturi meter offers a satisfactory method of measuring water, and is very often installed in pumping plants as it permits of the measurement of water without any interference in its flow. Figure 63 illustrates the arrangement for testing by the Venturi meter.

Figure 64 illustrates an arrangement for measuring the quantity of discharge by means of a Pitot tube. The tube measures the velocity head at the orifice of the discharge nozzle

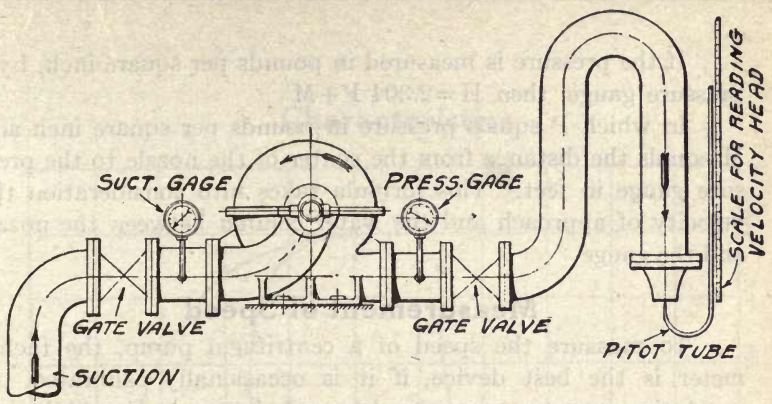


Fig. 64.

and the quantity of discharge can be found from the formula:

$$Q = CA\sqrt{2gH} \qquad (25)$$

In which Q equals the quantity of discharge in cubic feet per second, C is a constant for the nozzle which varies from .95 to .98. A equals the area of the nozzle in square feet, H equals the velocity head in feet.

The Pitot tube readings can only be accurate with a carefully calibrated nozzle and a careful determination of the coefficient C.

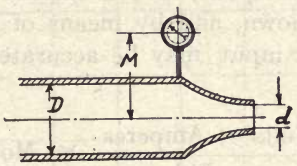


Fig. 65

Figure 65 illustrates a nozzle sometimes employed in the measurement of water. The nozzle used should be carefully calibrated, and a curve of discharge plotted, giving the quantity of discharge from the nozzle for various pressures at the point where the pressure gauge is attached. The formula for the discharge is

$$Q = 1963\, d^2 C \sqrt{\frac{H}{1 - C^2\left(\frac{d}{D}\right)^4}} \qquad (26)$$

In which C equals the coefficient of discharge which equals approximately .99, H is the pressure head in the nozzle in feet, d equals the diameter of the throat of the nozzle in inches, and D equals the diameter of the nozzle in inches at which pressure is measured.

If the pressure is measured in pounds per square inch, by a pressure gauge, then $H = 2.304\ P + M$

In which P equals pressure in pounds per square inch and M equals the distance from the center of the nozzle to the pressure gauge in feet. This formula takes into consideration the velocity of approach and the water column between the nozzle and the gauge.

Measurement of Speed

To measure the speed of a centrifugal pump, the tachometer is the best device, if it is occasionally calibrated. A revolution counter may also be used, but care should be exercised in its use as it does not indicate fluctuations in speed, and unless the readings are extended over a sufficient period of time, it will not give an average.

Measurement of Power

The horse power applied to the pump shaft may be measured in various ways depending upon the type of prime mover employed.

In testing laboratories, centrifugal pumps are tested by means of variable speed, direct current motors. The motor efficiencies are known, and by means of Volt Meters and Ammeters the power input may be accurately determined by the formula

$$\text{B. H. P.} = \frac{\text{Volts} \times \text{Amperes}}{746} \times \text{Motor Efficiency} \qquad (27)$$

The transmission dynamometer is also used, and is a very accurate device. For alternating currents, the following formula is used for arriving at the brake horse power

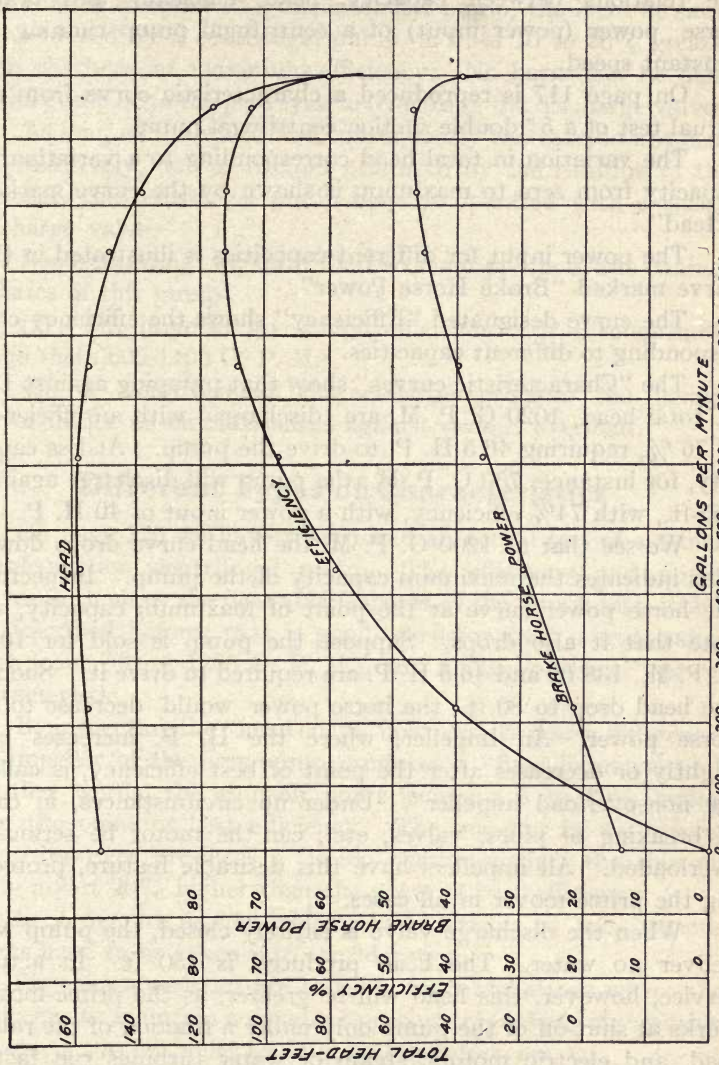
$$\text{B. H. P.} = \frac{\text{Volts} \times \text{Amperes} \times \text{Cos}\,\phi \times \sqrt{N} \times M}{746} \qquad (28)$$

M = Motor Efficiency.
N = Number of Phases.
Cos ϕ = Power Factor of Motor.

In the steam-engine driven pump, the power input is arrived at by indicator cards.

In the gas-engine driven unit, the power input is measured by the Prony brake.

Characteristics



Characteristics of a 5″ Double-Suction Pump Running at 1700 R. P. M. Constant Speed.

Fig. 66

On the preceding pages we state how the tests of centrifugal pumps are made. The results of these measurements can be produced graphically and the curves obtained illustrate best the relations between capacity, head, efficiency and brake horse power (power input) of a centrifugal pump running at constant speed.

On page 117 is reproduced a characteristic curve from an actual test of a 5″ double suction centrifugal pump.

The variation in total head corresponding to a variation in capacity from zero to maximum is shown by the curve marked "Head".

The power input for different capacities is illustrated in the curve marked "Brake Horse Power".

The curve designated "Efficiency" shows the efficiency corresponding to different capacities.

The "Characteristic curves" show that pumping against 138 ft. total head, 1020 G. P. M. are discharged with an efficiency of 76 %, requiring 46.5 H. P. to drive the pump. At less capacity, for instance, 750 G. P. M., the pump will discharge against 155 ft., with 74% efficiency, with a power input of 40 H. P.

We see that at 1200 G. P. M. the head curve drops down. This indicates the maximum capacity of the pump. Inspecting the horse power curve at the point of maximum capacity, we note that it also drops. Suppose the pump is sold for 1020 G. P. M., 138 ft. and 46.5 H. P. are required to drive it. Should the head drop to 80 ft., the horse power would decrease to 39 horse power. An impeller, where the H. P. increases but slightly or decreases after the point of best efficiency, is called "a non-overload impeller". Under no circumstances, in case of breaking of pipes, valves, etc., can the motor be seriously overloaded. All impellers have this desirable feature, protecting the prime-mover in all cases.

When the discharge valve is entirely closed, the pump will deliver no water. The head produced is 150 ft. In actual service, however, this head will be greater, as the prime-mover works at shut-off of the pump only under a fraction of the rated load, and electric motors, steam or water turbines run faster at a decreased load, thus increasing the pressure. The "Shut-off" point therefore will be the highest point on the head curve, which is of importance for installations where there is no friction head, and pump has to start against a maximum static head from

the beginning. To rotate the impeller at this point, 13 H. P. are required, representing the friction of rotation. As no useful work is performed at the shut-off (the pump delivering no water), the efficiency is zero. Special attention has to be called to the fact that in contrast to a displacement pump, the head at shut-off produced by a centrifugal pump, is from 10 to 20% greater than the head at maximum efficiency. No harm can be done to the pump or to the pipe system by closing the discharge valves.

As long as the water and consequently the case do not heat up excessively, due to friction produced by the rotation of the impeller, the centrifugal pump may be operated with a closed discharge valve.

Recapitulating, we state the two main points of the characteristics of this pump:

(1) The impeller gives high efficiency over a remarkable range (here 600-1100 G. P. M.).

(2) A non-overload characteristic protects the prime-mover under all circumstances against serious overload.

Different Types of Characteristics

On page 120 we have reproduced an actual test of a 6-inch double-suction centrifugal pump. The efficiency and power curves both show the desirable features of the 5-inch test, which we described before; that is good efficiency over a wide range (here from 700 to 1200 G. P. M.), and a "non-overload" power characteristic.

By inspecting the "head curve" however, we note a difference: the impeller of the first pump produces a "flat characteristic," in other words, the shut-off point is about 5 to 10% higher than the point of best efficiency. The impeller in the second pump however produces a "steep characteristic," the shut-off being about 20% higher than the point of best efficiency.

In order to decide which characteristic to use, various points have to be taken into consideration.

Flat characteristics are suitable for installations where the pump has to maintain a constant pressure regardless of variation in capacity, as for boiler feeding, accumulators, etc.

Steep characteristics are to be used for installations pumping against variable heads, as pipe line pumps, or pumps where the head is purely composed of friction. In such a case, a steep characteristic will be safer, as should the head vary, a pump

with a steep characteristic will adjust itself easier than a pump with only a small variation in the head curve.

For every pump we have a set of different impellers to suit the different conditions of service. When ordering, state which kind of characteristic is desired, if steep or flat. This will enable us to produce a pump which will give the best service.

Characteristics of a 6″ Double-Suction Pump

Running at 1700 R. P. M. Constant Speed.

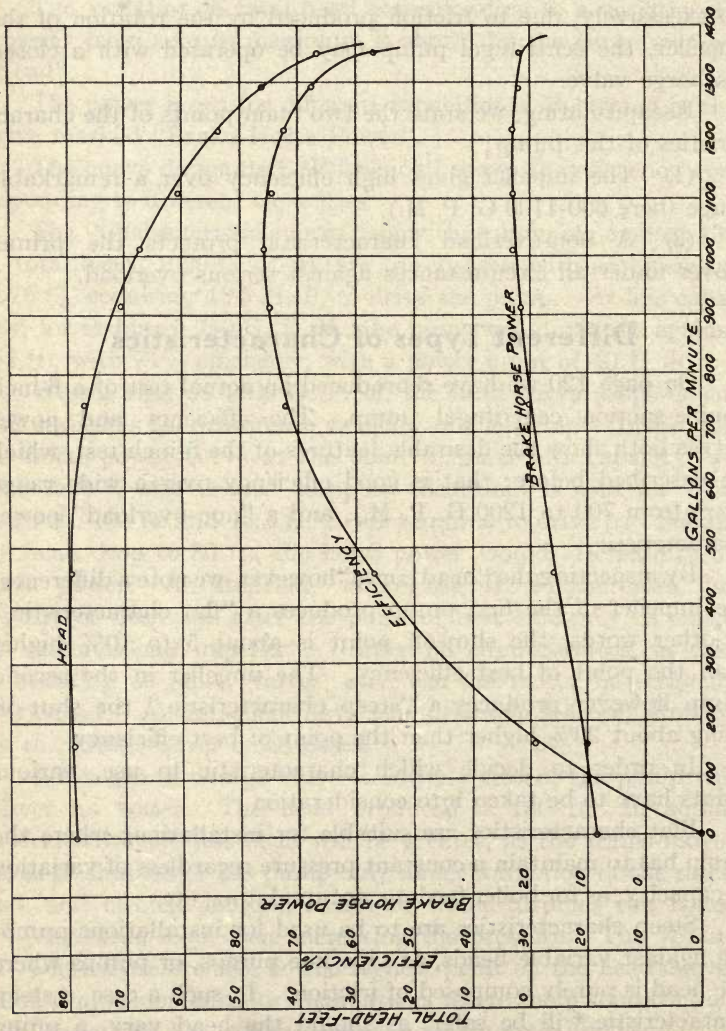


Fig. 67.

Efficiency of Centrifugal Pumps

The efficiency of a centrifugal pump is the percentage of work performed for the power delivered to the pump shaft, and is equivalent to the theoretical horse power divided by the horse power applied to the pump shaft.

The efficiency of a centrifugal pump varies according to the conditions of operation, but the following table will give a general idea of the efficiencies obtained in Union centrifugal pumps.

SIZE OF PUMP	Single-Stage Single-Suction Volute Pumps	Single-Stage Double-Suction Horizontal Pumps	Multi-Stage Pumps
¾"	20–25	.....	.....
1 "	20–30	.....	.....
1¼"	25–35	.....	.....
1½"	25–40	.....	40–53
2 "	30–50	40–60	45–59
2½"	30–52	.....	.....
3 "	45–56	48–73	45–60
4 "	50–65	60–78	60–67
5 "	55–68	60–80	63–70
6 "	55–70	65–81	64–73
8 "	65–72	65–82	65–76
10 "	65–73	67–83	68–79
12 "	66–75	68–84	.....
14 "	66–76	68–85	.....

Capacity, Speed, Head and Horse Power of Centrifugal Pumps

In case it is desired to alter the capacity, speed, or head of a centrifugal pump in order to suit some particular requirements, it is interesting to know that the capacity, speed, head, and horse power bear the following relations to each other.

Assuming that the quantity of water delivered by a centrifugal pump operating at N revolutions per minute is Q gallons per minute, and the total head the pump is operating against is H feet requiring P brake horse power to operate.

Now, if it is desired to increase the speed of this same pump to N_1 revolutions per minute, this will mean the pump will now deliver Q_1 gallons per minute, and the total head will be H_1 feet requiring P_1 horse power to operate under these conditions.

In order to simplify the equation, all constants are eliminated, as the results obtained are near enough for practical purposes.

$$\frac{Q}{Q_1}=\frac{N}{N_1} \quad (29) \qquad \frac{H}{H_1}=\frac{(N)^2}{(N_1)^2} \quad (30) \qquad \frac{P}{P_1}=\frac{(N)^3}{(N_1)^3} \quad (31)$$

In order to clearly illustrate the calculation of these formulae, we will assume that a centrifugal pump at its most economical point discharges 5000 gallons per minute against a total head of 50 feet, when operating at a speed of 900 revolutions per minute, and it requires 90 brake horse power. Now it is desired to increase this speed to 1000 R. P. M. How much water will this pump discharge at its most economical point and what is the total head and horse power required to operate this pump under these conditions?

Referring to the above formulae, and substituting the quantities therein, we have

Q ....5000 Gallons per minute.
N.... 900 Revolutions per minute.
H.... 50 Feet.
P .... 90 Brake horse power.
Q_1.... X Gallons per minute.
N_1....1000 Revolutions per minute.
H_1.... Y Feet.
P_1.... Z Brake horse power.

By Formula (29)

$$\frac{5000}{Q_1}=\frac{900}{1000} \qquad Q_1=\frac{5000\times 1000}{900}=5555=X \text{ Gals. per minute}$$

By Formula (30)

$$\frac{50}{H_1}=\frac{(900)^2}{(1000)^2} \qquad H_1=\frac{50\times(1000)^2}{(900)^2}=61.72=Y \text{ feet}$$

By Formula (31)

$$\frac{90}{P^1}=\frac{(900)^3}{(1000)^3} \qquad P_1=\frac{90\times(1000)^3}{(900)^3}=124=Z \text{ Brake H-Power}$$

If it is desired to keep the revolutions per minute constant, and alter (Q), (H) and (P), this may be accomplished within certain limits by changing the diameter of the impeller.

Assume D is the original diameter of the impeller (in inches or feet) and D_1 the proposed diameter of the impeller (in inches or feet), and substituting these factors in place of N and N_1 respectively in equations 29-31 inclusive, Q_1, H_1, and P_1 may be easily calculated.

In varying the conditions by changing the diameter of the impeller, it should be borne in mind that the pump casing determines the maximum size of the impeller that can be used, also, that the diameter of the impeller cannot be reduced beyond a certain figure on account of the excessive losses between the impeller and casing.

Prime Movers For Driving Centrifugal Pumps

The majority of centrifugal pumps in service are motor driven. Care should be exercised in choosing the size and type of motor to be used. The exact capacity and rating of the motor is of great importance, especially where the power is bought on the motor rating. If the motor is too small, it is liable to be overloaded, and if it is too large the customer pays for excess power.

In choosing a motor, the maximum conditions should be considered, and the power required at rated speed and head should not in itself determine the size of the motor.

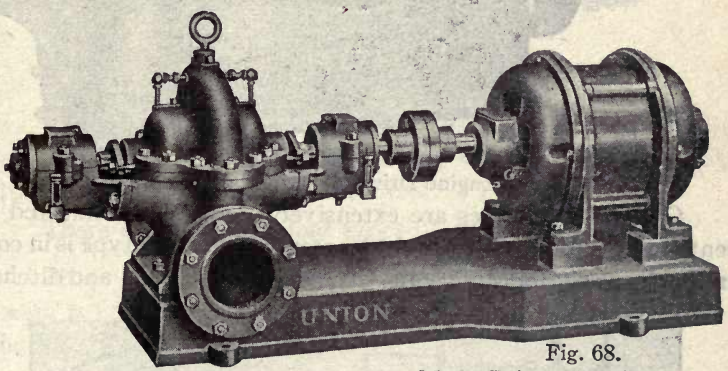


Fig. 68.
Motor-Driven Centrifugal Pump.

Having determined the size of the motor to be used, the next important problem is to select one of proper type for the service. If the head must vary, this can be accomplished by changing the speed of the motor, and a variable speed motor must be selected.

With centrifugal pumps, squirrel cage motors are generally used in the alternating current type, and compound wound motors in the direct motor type.

Steam Engine and Miscellaneous

Since the development of high speed steam engines, more engine-driven centrifugal pumps have been used. This type

of installation is desirable in small plants where the engineer is more familiar with the engine, than the motor, or the steam turbine. Steam engine driven centrifugal pumps are generally low head pumps, and are mostly used for circulating pumps.

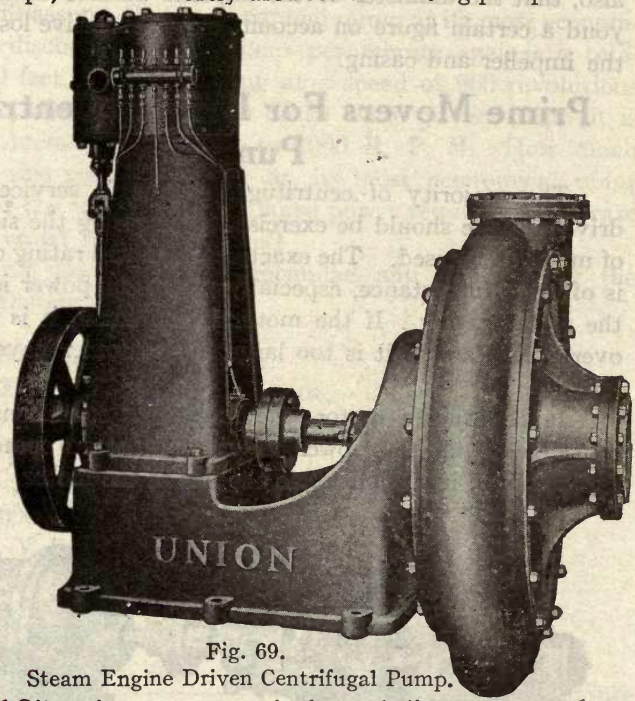


Fig. 69.
Steam Engine Driven Centrifugal Pump.

Gas and Oil engines are extensively used direct connected to centrifugal pumps. One of the largest fields for this type is in contractor's work for emptying excavations, sumps, and ditches.

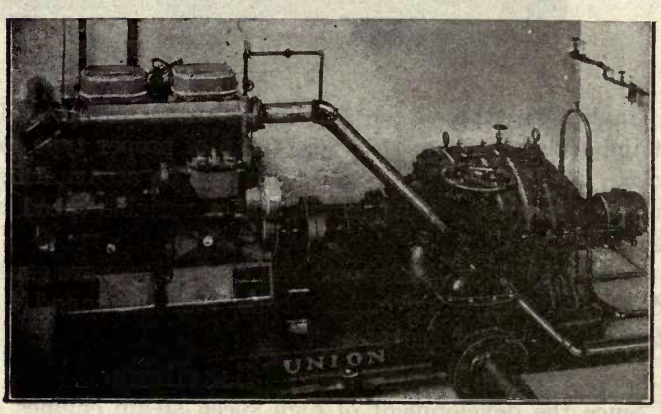


Fig. 70.
Gas Engine Driven Centrifugal Pump.

Steam Turbines

Centrifugal pumps driven by steam turbines are used extensively for boiler feed, hot well and circulating pumps. These installations are very efficient, compact and require a minimum amount of attention from the operator.

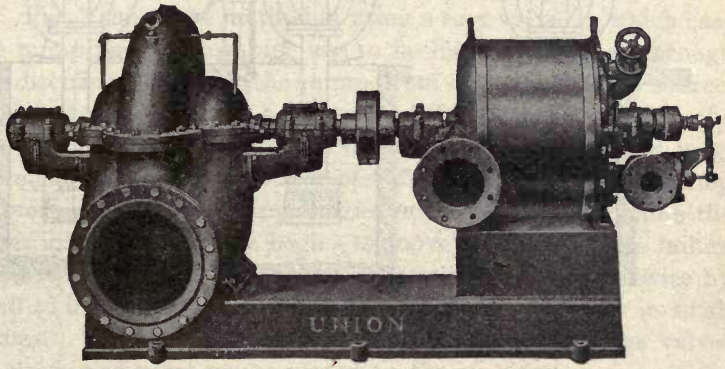


Fig. 71.
Steam Turbine Driven Centrifugal Pump.

In steam turbine driven units, the turbine may be direct connected or connected to the pump shaft by reducing gears, either of which is an economical drive.

There are four principal types of steam turbines.

First: The De Laval, an impulse turbine in which the steam is completely expanded in a single set of nozzles, and all the kinetic energy is given up in a single row of blades.

Second: Parsons, an impulse-reaction type where the energy of reaction of an expansion in the moving blades is added to the impulse of the steam as received from the fixed nozzles.

Third: Zoelly & Rateau impulse type having a series of partial expansions, the energy of each expansion being absorbed in a single row of moving blades.

Fourth: Curtis, in which the velocity of steam from the nozzles is absorbed in and passes through several rows of moving blades.

Methods of Priming Centrifugal Pumps

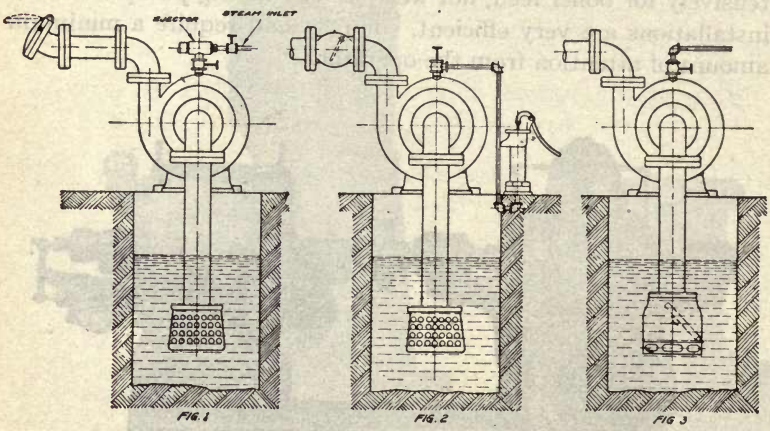


Fig. 72.

In the cuts above are shown three methods of priming. Fig. 1 shows a horizontal pump fitted with a discharge valve and a steam ejector. The discharge valve being closed, the steam inlet valve to the ejector being opened first, and then the valve between the ejector and the pump opened, the air in the pump and pipes will be exhausted, and the water drawn up into them. When the ejector is placed near the pump, complete priming will be indicated by water issuing from the ejector. In shutting off the ejector, close the valve between it and the pump first, and the steam inlet valve last. Where it is not convenient to place the ejector near the pump; the air pipe may be extended, in which case it is necessary that a slightly larger air pipe be used than when the ejector is placed near the pump.

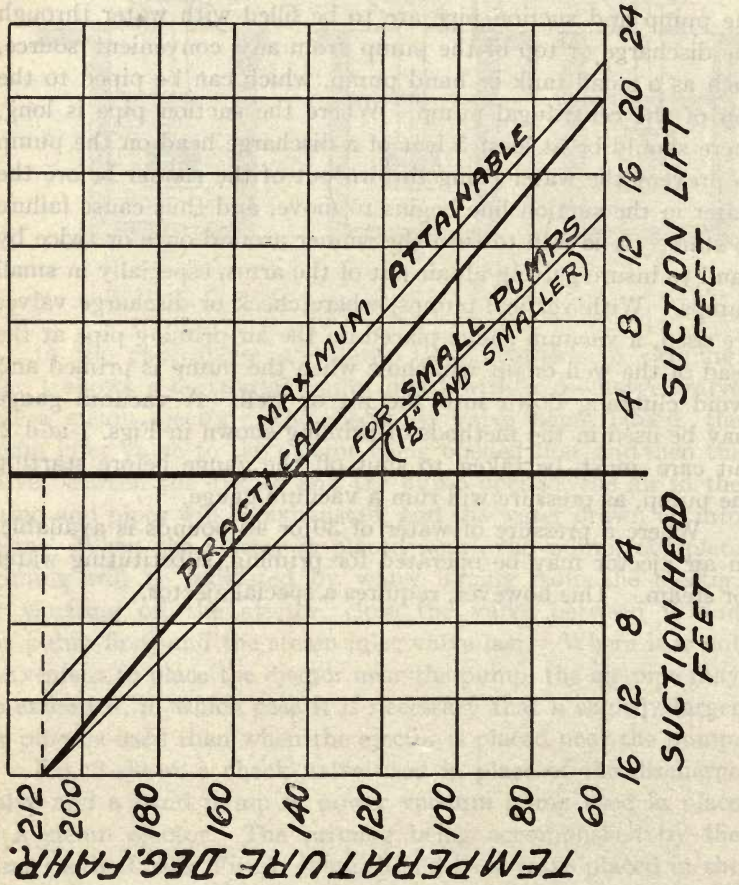
Fig. 2 shows a check valve used in place of the discharge valve and a hand pump or power vacuum pump used in place of a steam ejector. The priming being accomplished by the same method as in Fig. 1, there should be a valve placed in the air pipe which must be closed before starting. For hand priming, a common pitcher or Douglas pump may be used, piped as shown in the cut, that is, with the air pipe forming a loop a little above the discharge. It is only necessary to put a little water into this style of pump to water-seal it, and make of it a very good vacuum pump, the loop referred to above prevent-

ing the water from escaping. When a power driven vacuum pump is used for priming, a water trap or other means should be placed in the air pipe to prevent water entering and possibly breaking the primer. Where the valves on the vacuum pump are large, this may be omitted. On large pumps, a water glass similar to those used on boilers, placed near the top of the pump shell, will show when the priming is complete.

Fig. 3 shows the method of using a foot valve, in which case the pump and suction pipe are to be filled with water through the discharge or top of the pump from any convenient source, such as a small tank or hand pump, which can be piped to the top of the centrifugal pump. Where the suction pipe is long, there should be at least 5 feet of a discharge head on the pump to prevent the water being thrown out of the runner before the water in the suction line begins to move, and thus cause failure to start. It is well to turn the runner around once or twice by hand to insure getting all air out of the arms, especially in small pumps. With vertical pumps, where check or discharge valves are used, a vacuum gauge placed on the air priming pipe at the head of the well or pit will show when the pump is primed and avoid climbing down into the pit or well. A vacuum gauge may be used in the methods of priming shown in Figs. 1 and 2, but care must be taken to shut off the gauge before starting the pump, as pressure will ruin a vacuum gauge.

Where a pressure of water of 30 or 40 pounds is available, an air ejector may be operated for priming, substituting water for steam. This however, requires a special ejector.

Suction Lift—Head Diagram for Centrifugal Pumps
Fig. 73.



Directions for Installing Centrifugal Pumps

Set base-plate with pump and prime-mover on a solid foundation, preferably of concrete, and level carefully; grout the base-plate in and allow to set; see that pump and motor are in line, and shaft turns easily by hand after anchor-bolt nuts have been tightened.

When pump is set, connect suction and discharge pipes; see that these pipes are self-supporting, in order to avoid strain on pump and insure proper alignment.

Locate pump as near the water supply as possible. The maximum suction lift including pipe friction, varies with the size of pump, and the liquids to be handled. Hot water or heavy liquids must flow to the pump under a positive head, this head varying according to the difference in temperature of the liquid, or the viscosity of the liquid. The diagram on page 128 shows the lifts possible for different sizes of pumps with varying temperatures of the water.

Never use pipe sizes smaller than pump calls for. For long pipe lines, use 2 or 3 sizes larger. Make suction pipe as short as possible, and avoid bends and elbows. Make sure that there are no air pockets in the line and that the suction pipe is absolutely air tight.

Place a check valve and a gate valve in the discharge pipe, as close as possible to the pump. The check valve must be placed between the gate valve and the pump in order to be able to inspect or repair the check valve without being forced to empty the discharge line.

Always provide a strainer on the end of the suction line. This protects the pump from being choked by foreign matter, and insures safe operation.

To avoid corrosion, packing is removed from stuffing boxes before pump leaves factory. When repacking, do not draw the glands up tight, but allow a reasonable leakage, which lubricates the packing and prevents the shaft being corroded. Pipe the leakage from the tap in the bearing bracket to a sewer or drain.

Before starting pump, clean the bearings with gasoline or kerosene, and fill oil reservoirs with high grade lubricating oil, as high as the top of the oil gauge indicates.

Directions for Operating Centrifugal Pumps

The pump must always be primed before starting, otherwise the interior parts which depend upon the water for lubrication, will be injured. Never run a centrifugal pump empty.

The three methods of priming centrifugal pumps are fully described on pages 126-127.

As soon as case is primed by either method, pump should be started with discharge valve closed, and brought up to speed.

Then open the gate valve slowly until desired quantity of water is obtained. Failure of pressure to increase with the speed indicates air in the pump casing. In this case, stop the pump and prime again.

As long as the water and consequently the case do not heat up excessively, due to friction produced by the rotation of the impeller, the centrifugal pump may be operated with a closed discharge valve. In contrast with displacement pumps, no by-passes are required, nor can the pump or the pipe system be damaged as the shut-off pressure is only 10 to 15% greater than the pressure at full capacity.

Always run the pump in direction of arrow cast on case. Centrifugal pumps can be run only in one direction.

During the operation, stuffing boxes and bearings must be inspected occasionally. The centrifugal pump does not require any other attention.

If the pump is to be idle for long periods, it should be taken apart, cleaned and oiled. This prevents parts rusting together and preserves their good condition.

If pump is exposed to freezing temperature, it should be drained immediately after stopping.

These suggestions are intended to assist in installing and operating Union Centrifugal pumps. In all cases, successful operation depends largely on correctness of installation, for which this Company cannot be held responsible. For further information regarding Union Centrifugal Pumps, write us.

Some Centrifugal Pumps "Ifs"

If after starting the pump it throws a little water at the first few revolutions, and then churns and fails to discharge more, it is just evidence that the air was not all out of the pump and pipes, or the suction lift too great, or a leaky pipe, or a long suction and insufficient discharge head.

If when first started the pump throws a full stream for a few minutes, and then fails, it is caused by failure of supply, or water receding in the well below the suction limit, which in a well is best determined by a vacuum gauge placed on the suction elbow of the pump. The remedy for this is to lower the pump, thus reducing the suction lift.

If the pump delivers a full stream of water at the surface, or level of the pump, but fails to pump at a higher discharge point, the speed of the pump is too low.

If the pump starts a full stream, and then the discharge decreases very slowly until the pump fails to deliver any water, it is caused by an air leak at the packing gland.

If the pump delivers a full quantity for a few hours and fails, the speed and water supply being unchanged, the suction pipe or impeller is obstructed.

If when running there is a heavy vibration, the shaft has been sprung, the pump is out of alignment, or an obstruction has lodged in one side of the impeller.

If the bearings heat unduly, the belt is unnecessarily tight, the bearings lack oil, or there is an end thrust.

The last "If" is, that if the pump is properly installed, and operating, it will positively operate successfully, as the centrifugal is the most simple, most efficient and long lived water lifting device manufactured.

In pumping hot water or fluids, the suction lift of the pump must be as small as possible on account of the lowering of the boiling point under vacuum, and consequent loss of priming from the presence of vapor.

Water should not discharge into a sump or tank near the end of the suction pipe, as there is danger of carrying air down, and into the suction pipe.

Do not attempt to pump more than the maximum catalogue rating of the pump, as that will cause waste of power.

Do not call for help until you are sure none of the above "If's" are present.

To Calculate Horse Power

To determine the theoretical horse power to elevate a given quantity of water to a given height, multiply the number of gallons delivered per minute by 8.33 (weight of one gallon of water), multiply this result by the total head, and divide this result by 33000 (33000 pounds elevated one foot in one minute equals one horse power). This formula is the theoretical horse power, and when simplified is:

H. P. = .000252 x Gallons per minute × Head in feet. (32)

To determine the actual, or brake horse power, divide the theoretical, or water horse power by the efficiency of the pump.

EXAMPLE: It is desired to elevate 200 gallons of water per minute, to an elevation of 150 feet through 200 lineal feet of 3″ pipe with three 3″-90° elbows, and one 3″ globe valve.

The friction loss of 200 gallons per minute through 200′ of 3″ pipe from the table, on page 145 = 11.54 × 2 = 23.08′.

The friction loss of 200 gallons per minute through three 3″-90° elbows from the table, on page 147 = 1.18 × 3 = 3.54′.

The friction loss of 200 gallons per minute through one 3″ globe valve from the table on page 148, is equivalent to 24 lineal feet of 3″ straight pipe, and from the table on page 145, the friction loss of 200 gallons per minute through 24′ of 3″ pipe $= 11.54 \times \frac{24}{100} = 2.76'$.

The total head, therefore is equal to the sum of the above heads or

Friction head in pipe	= 23.08′
Friction head in elbows	= 3.54′
Friction head in valve	= 2.76′
Static Head	= 150.00′
Total head	= 179.38′

The theoretical or water horse power, equals from formula (32)

.000252 × 200 × 179.38 = 9.1 = Water horse power.

Assuming an efficiency for the pump of 55%, the actual or brake horse power necessary to operate the pump =

$$\frac{9.1 \times 100}{55} = 16.5 \text{ Brake horse power.}$$

So, in choosing a motor for these conditions, it would be advisable to use a 20 horse power motor, as the nearest size, which is 15 horse power, is too small, and would be overloaded. The 20 horse power motor allows a margin for unforeseen future changes in operating conditions, which might increase the load.

Cost of Pumping

The total cost of pumping is the sum of the operating expenses, and the fixed charges. The former consists of labor, fuel, electric current, supplies, etc. The latter consists of interest on the first cost, insurance, taxes, depreciation and administration.

The first cost covers the cost of pumping equipment. The total annual cost consists of fixed charges and operating costs for a year. The cost of pumping per water horse power, per 1000 gallons per minute, or any similar unit, is the total annual cost divided by the total capacity of the pump in these units.

It is a minimum when the pump is not operated, as it will consist only of fixed charges, and is a maximum when the pump is running continuously, as it will make the operating expenses a maximum.

On a steam driven pumping unit, such as a steam turbine, or engine driven pump, or a fly wheel pumping engine

$$T = \frac{Q \times 8.33 \times H \times C}{D} + L + F(i + d + t) + a \quad (33)$$

in which

T = Total Annual Cost
Q = Total number of gallons pumped per year.
H = Head in feet.
C = Cost of steam per 1000 lbs.
D = Duty in foot pounds for 1000 lbs. of steam.
L = Labor cost, etc.
F = Total investment
i = Interest rate on the investment.
d = Rate of depreciation.
t = Taxes and insurance.
a = Administration costs.

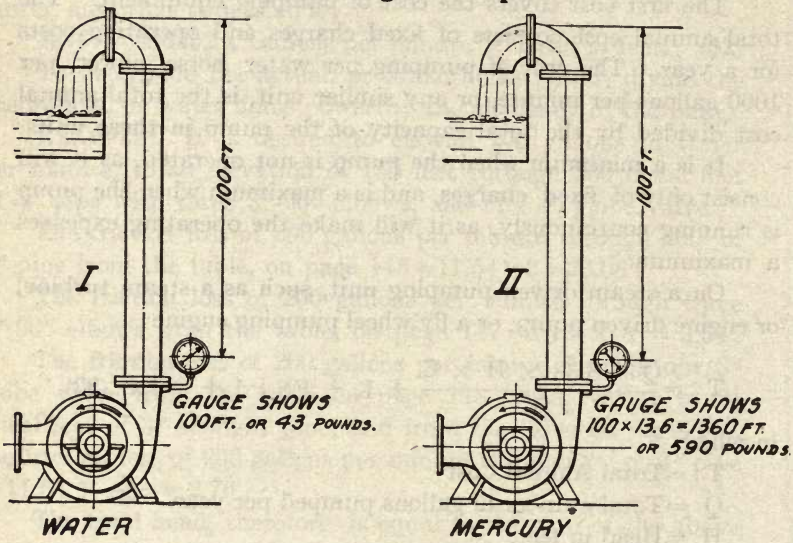
In an electric driven pumping unit, it is necessary to consider the cost per 1,000,000 B. T. U. supplied the motor instead of C., in the above equation. (1 K. W. Hr. = 3412 B. T. U.) Then D. = Duty in foot pounds per 1,000,000 B. T. U. supplied to the motor.

Since 1 B. T. U. = 778 foot pounds, 1,000,000 B. T. U. = 778,000,000 foot pounds, and Duty = 778,000,000 × efficiency of the unit. Therefore, the above equation transformed on this basis, assuming K = the cost of the electricity per K. W. Hr., is

$$T = \frac{Q \times 8.33 \times H \times K}{2{,}655{,}000 \times \text{efficiency of unit.}} + L + F(i + d + t) + a \qquad (34)$$

From the above equation, it is apparent that if the cost of power is high, it is an economical investment to buy an efficient pump.

Pumping Liquids Other Than Water With Centrifugal Pumps



In order to make a drastic comparison, assume two identical pumps, discharging under the same conditions and both running at the same speed. Pump No. 1 will handle water and pump No. 2 will handle mercury. The static head in both cases will be 100 feet. The pumps are started, and it is observed that both the water column and the mercury column will stand exactly at the same height in the standpipe. This is explained by the fact that the head created by a centrifugal pump is not a pressure head, but a velocity head. This velocity head depends entirely on the velocity of the water, which velocity in turn was produced by the speed of the impeller. As both pumps run at the same speed, both liquids regardless of their specific gravity or weight will stand at the same height in the standpipe.

The difference in the weight of the liquids is shown by the pressure gauges. Let us assume that the pipe line is very large, so that pipe friction can be neglected. Both gauges will show the pressure of the column of liquid in the respective pipes. The gauge No. 1 will read 100 ft, or 43 lbs., which is the weight

per square inch of water column 100 ft. high. As mercury is 13.6 times heavier than water, gauge No. 2 will show 13.6×100 1360 ft. or 590 lbs. pressure.

The power required by these two identical pumps, handling different liquids, can easily be computed, considering that regardless of the method, whether the head is obtained by centrifugal force or by piston pressure, the power required for pumping any kind of liquid is always the product of the weight of the liquid and the height it has to be elevated. Therefore, should the pump No. 1, which handles water, require 10 H. P., the pump No. 2 will require $13.6 \times 10 = 136$ H. P., as both pumps work against the same head, with the only difference that No. 2 handles a liquid 13.6 times heavier than No. 1.

Nothing has been said about the volume of the liquid handled. As both pumps have the same dimensions, it is self-evident that they will deliver the same volume of different liquids, as long as the viscosity does not enter into the question.

Recapitulation: In summing up we find:

(1) Regardless of the specific gravity of the liquid, a centrifugal pump will always produce the same static head.

(2) The pressure (read on the pressure gauge at the discharge of the pump) will be increased in proportion to the specific gravity.

(3) The horse-power required by the pump will also be increased in proportion to the specific gravity.

Examples: (1) Suppose a pump is required to elevate brine of 1.2 specific gravity to a static head of 100 ft. As explained before, any pump suitable for lifting water to a static head of 100 ft., will also lift brine to the same height. The pressure gauge will register the weight of a column of brine 100 ft. high. A column of water 100 ft. high would show 43 lbs. pressure. As brine is 1.2 times heavier than water, the gauge will show $43 \times 1.2 = 52$ lbs. pressure.

Assuming that it required 10 H. P. to drive the water pump, the brine pump will require $1.2 \times 10 = 12$ H. P.

(2) Suppose a pump is required to pump the same brine of 1.2 specific gravity against a pressure of 43 lbs. This time 43 lbs. are required on the pressure gauge of the brine pump. But if a water pump of proper proportions to discharge against 100 ft. or 43 lbs. is used, the gauge would show 52 lbs. when handling brine, as explained before. Therefore, to reduce this pressure to the required 43 lbs., it is necessary to change the impeller accordingly.

As seen in example No. 1, it required 10 H. P. to work against 43 lbs. water pressure. Naturally it takes the same 10 H. P. to pump against 43 lbs. brine pressure.

In most cases the brine pumps are used as circulating pumps and work against a balanced head; in other words, the pump has only to overcome the pipe friction. Therefore the head for such a case has to be specified as a pressure head, and not as a static head. This is insignificant for small installations; for larger installations, however, it will mean using a smaller motor (for instance 50 H. P. instead of 75 H. P).

(3) Suppose a pump is required to deliver gasoline of 0.8 specific gravity against a static head of 100 ft.

Figured for water, the pump will require say 10 H. P. As the pump will have to work against a static head of 100 ft., the pressure gauge will read only 80 ft. (corresponding to a specific gravity of 0.8), and the pump will require $10 \times 0.8 = 8$ H. P. only.

(4) Suppose a pump is required to handle gasoline of 0.8 specific gravity against a pressure of 43 lbs.

This time 43 lbs. are required on the pressure gauge of the pump; therefore, if the gasoline pump is designed like the water pump, only 34 lbs. would be shown by the pressure gauge, corresponding to the weight of the gasoline column. Therefore, in order to obtain 43 lbs. pressure with gasoline, it is necessary to increase the impeller sufficiently to give 43 lbs. pressure on the gauge. This pump, of course, will require 10 H. P., exactly the same amount as required for water pumping against 43 lbs.

As to liquids of thick consistency, regardless of their specific gravity, it is general experience that, compared with water, a pump delivers less capacity and requires more horse-power to drive it. This is explained by the fact that the thick liquid produces more skin friction and therefore offers more internal resistance to the moving parts of the pump. No general rule has been established yet, and in cases of doubt, experiment is the only way to find out the conditions. Therefore, should any cases arise where thick liquids are to be handled, it is necessary to provide ample power. In chemical plants, sugar mills, etc., it is general practice to heat heavy liquids so they will flow freely and then pump them with centrifugal pumps. In such cases the liquid should always flow to the pump under a suction head. This is also necessary for hot water, acids and for any liquids where vaporization is liable to occur.

The Materials Used for Pumping Various Liquids

The following table gives the proper materials to be used in centrifugal pumps for handling different kinds of liquids.

Kind of Liquid	Chemical Symbol	Material Used
Acetic Aid, Concentrated	$C_2H_4O_2$	All bronze fitted
Acetic Acid, Diluted	$C_2H_4O_2+H_2O$	All bronze fitted
Acid Mine Water		All bronze fitted
Alkaline Water		All iron fitted
Alcohol (crude)		Bronze fitted
Ammonia Water (Aqua Ammonia)	NH_3+H_2O	All iron fitted
Anilin Water		All iron fitted
Benzene	C_6H_6	All iron fitted
Benzine		Bronze fitted
Beer		Bronze fitted
Beer-wort		Bronze fitted
Beet Juice (thin)		All iron fitted
Bisulphite (all percentages)		Bronze fitted
Bitter Mineral Water		All bronze fitted
Brine (calcium)		All iron fitted
Brine (sodium)		Bronze fitted
Cane Juice		Bronze fitted
Carbonate of Soda	Na_2CO_3	All iron fitted
Caustic Carbonate of Soda in Solution, Hot		All iron fitted
Caustic Chloride of Magnesium Solution, Cold	Neutral	All iron fitted
Caustic Cyanogen in Solution	$CNOH+H_2O$	All iron fitted
Caustic Manganese in Solution	$MnOH+H_2O$	All iron fitted
Caustic Potash Solution	$KOH+H_2O$	All iron fitted

(Continued on next page.)

The Materials Used for Pumping Various Liquids (*Continued*)

Kind of Liquid	Chemical Symbol	Material Used
Caustic Potash Niter in Solution		All iron fitted
Caustic Soda Solution	$NaOH+H_2O$	All iron fitted
Caustic Sodium Chloride Solution	$NaClOH+H_2O$	All iron fitted
Caustic Zinc Chloride	$ZnClOH+HO$	All iron fitted
Chlorine	Cl	Attacks all metals
Chloride of Potash Solution	$KCl+H_2O$	All iron fitted
Caustic Chloride of Magnesium Solution, Hot		Hard lead
Coal Tar Oil		All iron fitted
Creosote Oil		All iron fitted
Cresol	$C_6H_4CH_3OH$	All iron fitted
Cyanogen	C_2N_2	Bronze fitted
Cyanide of Potassium	$KCN+H_2O$	All iron fitted
Distillery wort		All bronze fitted
Ferrous Chloride	$FeCl_4$	All iron fitted
Gasoline		Bronze fitted
Glycerin	$C_3H_8O_3$	All bronze fitted
Green Vitriol	$FeSo_4+7H_2O$	All iron fitted
Guncotton Brine		All iron fitted
Heavy Oil		Bronze fitted
Hydrochloric Acid in thin Solution		All bronze fitted
Hydrochloric Acid	HCL	All bronze fitted
Iron Pyritic Acid		All bronze fitted
Lime Water		All iron fitted
Linseed Oil		Bronze fitted
Lye (caustic)		All iron fitted
Lye (containing much salt)		Bronze fitted

(*Continued on next page*)

The Materials Used for Pumping Various Liquids—(*Continued*)

Kind of Liquid	Chemical Symbol	Material Used
Lye Solution containing sand		All iron
Mash		All bronze fitted
Milk		All bronze fitted
Mineral Oil		Bronze fitted
Molasses		Bronze fitted
Naptha		All iron fitted
Nitric Acid, Concentrated		Lead
Nitric Acid, Diluted		All iron fitted
Paraffin, hot	C_nH_{2n+2}	Bronze fitted
Petroleum		All iron fitted
Petroleum Ether		All iron fitted
Potash Solution	$K_2CO_3+H_2O$	All iron fitted
Pulp		Bronze fitted
Purifying Oil		All iron fitted
Rape Oil		Bronze fitted
Salt Water		Bronze fitted
Sea Water		Bronze fitted
Sewage		Bronze fitted
Sebacic Acid		All bronze fitted
Syrup		Bronze fitted
Soap Water		All iron fitted
Soda	NaOH	All iron fitted
Sodium Chloride Solution	$NaCl+H_2O$	Bronze fitted
Sodium Sulphate	Na_2SO_4	All iron fitted
Sugar Compound		Bronze fitted
Sugar Solution		Bronze fitted
Sulphate of Lime	$CaSO_4$	Bronze fitted

(*Continued on next page.*)

The Materials Used for Pumping Various Liquids—(*Continued*)

Kind of Liquid	Chemical Symbol	Material Used
Strontia in Caustic Solution	$SrOH+H_2O$	All iron fitted
Sulphide of Hydrogen	H_2S	All bronze fitted
Sulphite of Sodium, Hot		All iron fitted
Sulphuric Acid, Hot		All iron fitted
Sulphuric Acid, Concentrated		All iron fitted
Sulphuric Acid, Common	H_2SO_4	All bronze fitted
Sulphurous Acid, Concentrated	H_2SO_3	All bronze fitted
Sulphurous Acid, Diluted		All bronze fitted
Tannic Acid	$C_{14}H_{10}O_3$	All bronze fitted
Turpentine Oil		All iron fitted
Vegetable Oil		All iron fitted
Vinegar		All bronze fitted
Wine		All bronze fitted
Wood Alcohol	CH_4O	Bronze fitted
Water, (Hot or Cold)		Bronze fitted
Water Containing Sulphur		Bronze fitted
Water Containing some Tar and Ammonia		All iron fitted
Wood Pulp		Bronze fitted
Whiskey		All bronze fitted

ABBREVIATIONS

All iron fitted signifies: Iron case, iron impeller and steel shaft; iron diffusor for multi-stage pumps.

Bronze fitted signifies: Iron case, bronze impeller (with or without bronze wearing rings), and steel shaft (with or without bronze protecting sleeves); bronze diffusor for multi-stage pumps.

All bronze fitted signifies: Bronze case, bronze impeller (with or without bronze wearing rings), and steel shaft with bronze protecting sleeves; bronze diffusor for multi-stage pumps.

Consumption of Gasoline For Pumping

Consumption of Gasoline or Distillate per Effective Horse-power per Hour In Gallons.	Approximate Consumption of Gasoline or Distillate per 24 Hours when Pumping 100 Gallons per Minute 100 Feet Head. In Gallons.				
	Total Efficiency of Pump and Drive.				
	30 per cent.	40 per cent.	50 per cent.	60 per cent.	70 per cent.
1/10	20	15.	12.	10.	8.7
1/9	22.5	17.	13.6	11.3	9.6
1/8	25.5	19.2	15.2	12.8	10.9
1/7	29.	21.5	17.2	14.4	12.3
1/6	33.5	25.	20.	16.7	14.3
1/5	40.	30.	24.	20.	17.3

To find the approximate number of gallons of gasoline or distillate required per hour when pumping 600 to 1,200 gallons per minute, multiply the water horse power by .027, or when pumping 100 to 500 gallons per minute, multiply by .032.

The water horse power is found by multiplying the weight of water pumped per minute by the total height in feet, and dividing by 33,000.

Another method of calculating the theoretical water horse power is to multiply the gallons per minute by the total head in feet, including friction in the pipes, and dividing by 4,000.

Consumption of Electric Current For Pumping.

Total Efficiency of Pump, Motor and Transformer.	Consumption of Electric Current in Kilowatt Hours per 24 Hours Run, Pumping 100 Gallons per Minute 100 Feet High.	Total Efficiency of Pump, Motor and Transformer.	Consumption of Electric Current in Kilowatt Hours per 24 Hours Run, pumping 100 Gallons per Minute 100 Feet High.
Per Cent.	Kilowatt Hours.	Per Cent.	Kilowatt Hours.
30	149.2	45	100.
32	142.	50	89.
34	132.	55	81.
36	124.	60	75.
38	118.	65	69.
40	112.	70	64.
42	107.	75	59.

The approximate kilowatt hours are equal to the horse power consumed by the pump divided by 1.1.

Heads in Feet of Water with Equivalent Inches of Mercury, Pounds per Sq. In. and Velocity in Ft. per Sec.

Computed with following Constants: g=acceleration of gravity=32.16; w=weight of cubic foot of water=62.4; Specific gravity of mercury =13.6.

Head in Feet	Inches of Mercury	Pounds per Sq. In.	Velocity Ft. per Sec.	Head in Feet	Inches of Mercury	Pounds per Sq. In.	Velocity Ft. per Sec.
1	.883	.433	8.02	50	44.2	21.66	56.6
2	1.765	.866	11.34	51	45.	22.09	57.3
3	2.65	1.30	13.89	52	45.9	22.54	57.8
4	3.53	1.73	16.04	53	46.8	22.98	58.4
5	4.41	2.17	17.93	54	47.7	23.40	59.0
6	5.30	2.60	19.65	55	48.5	23.84	59.5
7	6.18	3.03	21.22	56	49.4	24.27	60.0
8	7.06	3.47	22.69	57	50.3	24.71	60.6
9	7.95	3.90	24.06	58	51.2	25.13	61.2
10	8.83	4.33	25.36	59	52.1	25.58	61.6
11	9.71	4.76	26.60	60	53.	26.00	62.1
12	10.6	5.19	27.78	61	53.8	26.43	62.6
13	11.5	5.63	28.92	62	54.7	26.87	63.2
14	12.35	6.06	30.01	63	55.6	27.31	63.7
15	13.25	6.49	31.06	64	56.5	27.74	64.2
16	14.10	6.92	32.08	65	57.3	28.19	64.7
17	15.00	7.36	33.07	66	58.2	28.60	65.2
18	15.9	7.79	34.03	67	59.2	29.03	65.6
19	16.78	8.23	34.96	68	60.	29.47	66.2
20	17.65	8.66	35.87	69	60.9	29.91	66.6
21	18.52	9.09	36.75	70	61.8	30.34	67.1
22	19.41	9.12	37.62	71	62.7	30.79	67.6
23	20.3	9.96	38.46	72	63.5	31.20	68.1
24	21.2	10.40	39.29	73	64.5	31.63	68.5
25	22.1	10.83	40.10	74	65.3	32.07	69.0
26	22.95	11.27	40.90	75	66.2	32.51	69.5
27	23.8	11.67	41.67	76	67.1	32.95	70.0
28	24.7	12.12	42.44	77	68.	33.39	70.5
29	25.6	12.56	43.19	78	68.9	33.81	71.0
30	26.5	13.00	43.93	79	69.75	34.24	71.4
31	27.4	13.42	44.66	80	70.6	34.67	71.8
32	28.25	13.87	45.37	81	71.5	35.10	72.2
33	29.1	14.30	46.07	82	72.4	35.53	72.7
34	30.	14.72	46.77	83	73.3	35.98	73.1
35	30.9	15.16	47.45	84	74.1	36.40	73.5
36	31.8	15.60	48.12	85	75.	36.83	74.0
37	32.65	16.02	48.79	86	75.9	37.28	74.5
38	33.6	16.47	49.44	87	76.8	37.71	74.8
39	34.4	16.90	50.09	88	77.7	38.14	75.3
40	35.3	17.33	50.72	89	78.6	38.78	75.7
41	36.2	17.77	51.35	90	79.5	39.00	76.2
42	37.1	18.20	51.98	91	80.3	39.42	76.6
43	37.95	18.62	52.59	92	81.2	39.84	77.0
44	38.8	19.07	53.20	93	82.	40.3	77.5
45	39.7	19.50	53.80	94	83.	40.7	77.8
46	40.6	19.93	54.40	95	83.8	41.2	78.1
47	41.5	20.36	54.98	96	84.7	41.6	78.6
48	42.4	20.80	55.57	97	85.6	42.1	79.0
49	43.3	21.24	56.14	98	86.5	42.5	79.5

Heads in Feet of Water with Equivalent Inches of Mercury, Pounds per Sq. In. and Velocity in Ft. per Sec.

(Continued)

Computed with following Constants: g=acceleration of gravity=32.16; w=weight of cubic foot of water=62.4; Specific gravity of mercury =13.6.

Head in Feet	Inches of Mercury	Pounds per Sq. In.	Velocity Ft. per Sec.	Head in Feet	Inches of Mercury	Pounds per Sq. In.	Velocity Ft. per Sec.
99	87.4	42.9	79.8	400	353.	173.3	160.1
100	88.3	43.3	80.2	410	362.	177.7	162.2
105	92.7	45.4	82.25	420	371.	182.0	164.4
110	97.1	47.6	84.2	430	379.5	186.2	166.0
115	101.5	49.8	86.0	440	388.	190.7	168.1
120	105.9	51.9	88.0	450	397.	195.0	170.0
125	110.3	54.1	89.8	460	406.	199.3	172.0
130	114.8	56.3	91.5	470	450.	203.6	174.0
135	119.	58.4	93.2	480	424.	208.0	176.0
140	123.5	60.6	95.0	490	432.	212.4	177.7
145	128.	62.7	96.5	500	441.	216.6	179.5
150	132.3	64.9	98.2	510	450.	220.9	181.0
155	136.8	67.1	100.0	520	458.	225.4	183.0
160	141.1	69.3	101.3	530	468.	229.8	184.4
165	145.7	71.4	103.0	540	477.	234.0	186.1
170	150.	73.6	104.7	550	485.	238.4	188.0
175	154.4	75.8	106.1	560	494.	242.7	190.0
180	159.	77.9	107.5	570	503.	247.1	191.6
185	163.2	80.1	109.0	580	512.	251.3	193.0
190	167.7	82.2	110.5	590	521.	255.8	194.9
195	172.	84.4	112.0	600	529.	260.0	196.2
200	176.5	86.6	113.3	620	548.	268.7	199.9
210	185.2	90.9	116.2	640	565.	277.4	201.4
220	194.	95.2	119.0	660	583.	286.0	206.0
230	203.	99.6	121.8	680	600.	294.7	209.0
240	212.	104.0	124.2	700	618.	303.4	212.0
250	226.	108.3	127.0	720	635.	312.0	215.0
260	229.5	112.7	129.2	740	653.	320.7	218.0
270	238.1	116.9	132.0	760	671.	329.5	221.0
280	247.	121.2	134.2	780	688.	338.1	224.0
290	256.	125.6	136.6	800	706.	346.7	226.5
300	265.	130.0	139.0	820	724.	355.3	230.0
310	274.	134.2	141.0	840	742.	364.0	232.0
320	282.	138.7	143.3	860	758.	372.8	235.0
330	291.	143.0	146.0	880	776.	381.4	238.0
340	300.	147.2	148.0	900	794.	390.0	240.5
350	309.	151.6	150.0	920	812.	398.4	243.3
360	318.	156.0	152.0	940	830.	407.0	246.0
370	326.5	160.2	154.0	960	847.	416.0	248.5
380	335.	164.7	156.1	980	865.	425.0	251.0
390	344.	169.0	158.2	1000	883.	433.0	253.8

Table of Friction of Water in Pipes

Giving the velocity in feet per sec., the friction head in feet, and friction loss in lbs. pressure per sq. in. for each 100 feet in length of pipe, for different sizes of clean iron pipe discharging given quantities per minute.

Gallons per minute	Velocity in feet per second	Friction Head in feet	Friction loss in lbs. per sq. in.	Velocity in feet per second	Friction Head in feet	Friction loss in lbs. per sq. in.	Velocity in feet per second	Friction Head in feet	Friction loss in lbs. per sq. in.	Velocity in feet per second	Friction Head in feet	Friction loss in lbs. per sq. in.
		¾″			1″			1¼″			1½″	
5	3.64	7.59	3.3	2.04	1.93	0.84	1.3	0.71	0.31	0.91	0.27	0.12
10	7.28	29.90	13.0	4.08	10.26	3.16	2.6	2.41	1.05	1.82	1.08	0.47
15	10.92	66.01	28.7	6.12	16.05	6.98	3.9	5.47	2.38	2.73	2.23	0.97
20	14.56	115.92	50.4	8.16	28.29	12.30	5.2	9.36	4.07	3.64	3.81	1.66
25	18.2	180.00	78.0	10.20	43.70	19.0	6.5	14.72	6.4	4.55	5.02	2.62
30				12.24	63.25	27.5	7.8	21.04	9.15	5.46	8.62	3.75
35		4″		12.28	85.10	37.0	9.1	28.52	12.4	6.37	11.61	5.05
40	1.04	0.1383	0.06	10.32	110.40	48.0	10.4	37.03	16.1	7.28	14.99	6.52
45	1.17	0.1615	0.07				11.7	46.46	20.2	8.19	18.74	8.15
50	1.3	0.208	0.09		5″		13.	57.27	24.9	9.1	23.00	10.0
60	1.56	0.3	0.13	0.88	0.1156	0.05	15.6	85.50	37.0	10.92	32.95	14.25
70	1.82	0.439	0.19	1.04	0.162	0.07	18.2	114.00	49.3	12.74	44.6	19.3
75	1.95	0.485	0.21	1.2	0.174	0.075	19.5	129.00	56.1	13.65	51.52	22.4
80	2.08	0.581	0.23	1.28	0.185	0.08				14.56	58.45	25.3
90	2.34	0.6	0.26	1.44	0.208	0.09		6″		16.38	81.5	35.25
100	2.60	0.763	0.33	1.6	0.277	0.12	1.14	0.115	0.05		89.70	39.0
125	3.25	1.13	0.49	2.	0.393	0.17	1.42	0.161	0.07		7″	
150	3.8	1.59	0.69	2.4	0.578	0.25	1.71	0.231	0.10	1.2	0.0924	0.04
175	4.45	2.146	0.93	2.8	0.785	0.34	2.	0.302	0.13	1.38	0.115	0.05
185	4.7	2.484	1.075	2.96	0.84	0.363	2.11	0.36	0.1559	1.55	0.13	0.0562
200	5.1	2.82	1.22	3.2	0.972	0.42	2.28	0.393	0.17	1.7	0.162	0.07
250	6.4	4.37	1.89	4.	1.504	0.65	2.8	0.601	0.26	2.1	0.2775	0.12
265	6.79	4.645	2.09	4.24	1.69	0.731	3.03	0.70	0.303	2.23	0.31	0.134
300	7.6	6.15	2.66	4.8	2.15	0.93	3.4	0.85	0.37	2.4	0.393	0.17
350	8.9	8.44	3.65	5.6	2.914	1.26	3.9	1.15	0.50	2.94	0.531	0.23
370	9.41	9.33	4.04	5.92	3.197	1.382	4.22	1.33	0.576	3.1	0.59	0.255
400	10.1	10,92	4.73	6.5	3.72	1.61	4.5	1.50	0.65	3.36	0.693	0.30
450	11.4	13.88	6.01	7.3	4.62	2.00	5.05	1.87	0.81	3.96	0.855	0.37
480	12.16	14.34	6.21	7.68	5.06	2.192	5.49	2.16	0.935	4.03	0.96	0.415
500	12.7	17.16	7.43	8.1	5.55	2.40	5.6	2.22	0.96	4.2	1.04	0.45
750		14″		12.15	11.28	4.875	8.4	5.11	2.21	6.3	2.34	1.03
1000	2.224	0.1432	0.062				11.3	8.98	3.88	8.4	4.16	1.80
1100	2.444	0.18	0.078				12.57	10.90	4.715	9.25	5.08	2.20
1250	2.78	0.22	0.0952		15″		14.1	13.86	6.00	10.5	6.60	2.85
1500	3.336	0.306	0.135	2.7	0.22	0.0952		18″		12.6	9.45	4.08
2000	4.448	0.541	0.234	3.6	0.38	0.1645	2.74	0.195	0.0844			
2200	4.9	0.68	0.2943	3.96	0.455	0.197	3.18	0.26	0.1124			
2500	5.562	0.8365	0.362	4.5	0.575	0.248	3.425	0.295	0.1276		20″	
3000	6.674	1.19	0.515	5.4	0.79	0.342	4.11	0.405	0.175	3.308	0.25	0.108
3300	7.34	1.45	0.6285	5.94	0.93	0.402	4.52	0.475	0.2056	3.64	0.294	0.127
3500	7.786	1.61	0.697	6.3	1.045	0.452	4.795	0.53	0.229	3.86	0.325	0.1406
4000	8.898	2.10	0.910	7.2	1.445	0.675	5.48	0.68	0.294	4.41	0.415	0.1796
4500	10.0	2.65	1.15	8.1	1.67	0.723	6.165	0.84	0.363	4.955	0.51	0.2208
5000	11.1	3.3	1.43	9.	2.03	0.879	6.85	1.02	0.441	5.51	0.61	0.264
6000	13.4	4.8	2.07	10.8	2.9	1.256	8.22	1.425	0.616	6.61	0.87	0.3765
7300				13.13	4.16	1.804	10.	2.065	0.894	8.05	1.24	0.536
7500		30″		13.5	4.4	1.905	10.275	2.175	0.94	8.26	1.3	0.563
10000	4.77	0.315	0.137				13.70	3.67	1.588	11.02	2.245	0.972
10500	5.10	0.35	0.1515				14.4	4.16	1.8	11.58	2.47	1.07
14000	6.69	0.58	0.251		36″					15.43	4.25	1.84
15000	7.16	0.658	0.284	4.93	0.271	0.117						
17000	8.11	0.845	0.36	5.58	0.344	0.149		40″				
20000	9.55	1.13	0.489	6.57	0.471	0.204	5.28	0.22	0.0952			
25000	11.94	1.75	0.757	8.21	0.71	0.307	6.61	0.30	0.1298		45″	
30000	14.32	2.473	1.07	9.95	1.02	0.441	7.93	0.45	0.1946	6.255	0.35	0.1515
35000				11.59	1.37	0.593	9.25	0.63	0.2725	7.29	0.46	0.199
40000				13.23	1.765	0.764	10.59	0.81	0.35	8.33	0.585	0.253
45000				14.78	2.182	0.944	11.9	1.15	0.498	9.37	0.785	0.34
50000							13.22	1.6	0.6915	10.42	0.90	0.389
60000										12.5	1.273	0.551
70000										14.58	1.72	0.745
80000												
90000												
100000												

Table of Friction of Water in Pipes

(Continued)

Velocity in feet per sec., the friction head in feet, and friction loss in lbs. pressure per sq. in for each 100 feet in length of pipe, for different sizes of clean iron pipe discharging given quantities per minute.

Gallons per minute	2″ Velocity in feet per second	2″ Friction Head in feet	2″ Friction loss in lbs. per sq. in.	2½″ Velocity in feet per second	2½″ Friction Head in feet	2½″ Friction loss in lbs. per sq. in.	3″ Velocity in feet per second	3″ Friction Head in feet	3″ Friction loss in lbs. per sq. in.	3½″ Velocity in feet per second	3½″ Friction Head in feet	3½″ Friction loss in lbs. per sq. in.
5	0.49	0.092	0.04	0.244	0.046	0.02						
10	0.98	0.277	0.12	0.656	0.092	0.04	0.448	0.046	0.02			
15	1.47	0.577	0.25	0.985	0.185	0.08	0.672	0.092	0.04	0.498	0.046	0.02
20	2.04	0.97	0.42	1.315	0.323	0.14	0.896	0.138	0.06	0.664	0.069	0.03
25	2.6	1.43	0.62	1.645	0.485	0.21	1.12	0.231	0.10	0.83	0.092	0.04
30	3.03	2.09	0.91	1.97	0.693	0.30	1.345	0.30	0.13	0.996	0.138	0.06
35	3.54	2.76	1.22	2.29	0.92	0.40	1.569	0.393	0.17	1.163	0.208	0.09
40	4.05	3.68	1.60	2.62	1.19	0.53	1.79	0.53	0.23	1.329	0.254	0.11
45	4.56	4.60	1.99	2.95	1.49	0.66	2.016	0.647	0.28	1.494	0.323	1.14
50	5.1	5.61	2.44	3.3	1.86	0.81	2.24	0.80	0.35	1.66	0.393	0.17
60	6.12	8.88	3.50	3.95	2.7	1.17	2.688	1.155	0.50	1.992	0.555	0.24
70	7.14	11.09	4.80	4.6	3.46	1.50	3.136	1.385	0.60	2.324	0.879	0.38
75	7.7	12.23	5.32	4.93	4.14	1.80	3.360	1.70	0.75	2.490	0.913	0.395
80	8.16	14.55	6.30	5.26	4.62	2.00	3.584	2.08	0.90	2.656	0.948	0.41
90	9.18	18.02	7.80	5.91	5.96	2.58	4.032	2.54	1.10	2.988	1.247	0.54
100	10.2	21.75	9.46	6.5	7.36	3.20	4.480	3.01	1.31	3.320	1.478	0.64
125	12.8	34.27	14.9	8.13	11.24	4.89	5.6	4.57	1.99	4.15	2.219	0.96
150	15.3	48.76	21.2	9.8	16.10	7.00	6.8	6.55	2.85	4.98	3.12	1.35
175				11.43	21.75	9.46	7.92	8.85	3.85	5.81	4.208	1.82
185				11.84	24.5	10?61	8.34	9.94	4.3	6.14	4.62	2.00
200		8″		13.06	28.86	12.47	9.04	11.54	5.02	6.64	5.50	2.38
250	1.6	0.162	0.07				11.28	17.84	7.76	8.30	8.55	3.70
265	1.698	0.175	0.0757		9″		12.4	20.09	8.72	8.8	9.6	4.15
300	1.9	0.208	0.09	1.5	0.115	0.05	13.52	25.76	11.2	9.96	11.63	5.04
350	2.2	0.2775	0.12	1.75	0.1618	0.07				11.62	16.4	7.10
370	2.37	0.306	0.1324	1.85	0.178	0.077				12.28	18.13	7.85
400	2.6	0.37	0.16	2.	0.208	0.09		10″		13.28	21.36	9.25
450	2.92	0.462	0.20	2.25	0.2545	0.11	1.8	0.1618	0.07			
480	3.07	0.498	0.2155	2.4	0.3	0.13	1.92	1.1816	0.0785		12″	
500	3.2	0.5785	0.25	2.5	0.324	0.14	2.	0.204	0.09	1.4	0.0925	0.04
750	4.8	1.224	0.53	3.75	0.6945	0.30	3.	0.416	0.18	2.1	0.1848	0.08
1000	6.4	2.17	0.94	5.	1.224	0.53	4.	0.74	0.32	2.8	0.304	0.13
1100	7.04	2.64	1.142	5.5	1.50	0.649	4.4	0.88	0.381	3.08	0.375	0.162
1200	8.0	3.378	1.46	6.25	1.895	0.82	5.	1.132	0.49	3.5	0.462	0.20
1550	9.6	4.84	2.09	7.5	2.704	1.17	6.1	1.618	0.70	4.2	0.67	0.29
2000	12.7	8.70	3.765				8.1	2.842	1.23	5.6	1.132	0.49
2200	14.09	10.71	4.64				8.8	3.34	1.446	6.16	1.38	0.598
2500							10.1	4.43	1.92	7.	1.78	0.77
3000		22″					12.1	5.30	2.725	8.4	2.567	1.11
3300	2.973	0.18	0.078				13.2	7.50	3.24	9.24	3.08	1.333
3500	3.15	0.20	0.0865		24″		14.1	8.52	3.68	9.8	4.18	1.81
4000	3.6	0.255	0.114	3.08	0.175	0.0758		26″		11.3	4.45	1.925
4500	4.06	0.375	0.162	3.47	0.223	0.0965	2.874	0.14	0.0606	12.75	5.55	2.4
5000	4.5	0.40	0.173	3.855	0.27	0.117	3.19	0.22	0.0952	14.0	6.65	2.88
6000	5.4	0.55	0.234	4.62	0.375	0.162	3.815	0.241	0.104		28″	
7300	6.57	0.78	0.337	5.625	0.53	0.229	4.658	0.35	0.1515	4.015	0.235	0.1019
7500	6.75	0.825	0.357	5.783	0.56	0.232	4.785	0.363	0.157	4.125	0.252	0.1092
10000	9.	1.392	0.633	7.708	0.94	0.4065	6.38	0.62	0.2685	5.5	0.43	0.186
10500	9.45	1.53	0.663	8.1	1.03	0.445	6.7	0.68	0.2944	5.77	0.475	0.26
14000	12.6	2.64	1.142	10.79	1.792	0.776	8.93	1.15	0.498	5.7	0.82	0.3555
15000	13.5	3.01	1.304	11.56	2.05	0.888	9.57	1.305	0.565	8.25	0.93	0.405
17000	15.3	3.8	1.645	13.10	2.6	1.125	10.84	1.66	0.729	9.35	1.17	0.56
20000				15.4	3.52	1.524	12.76	2.125	0.97	11.00	1.6	0.693
25000										13.75	2.45	1.062
30000												
35000		50″			55″							
40000	6.76	0.358	0.155	5.56	0.226	0.081		60″				
45000	7.605	0.446	0.193	6.25	0.281	0.1216	5.245	0.187	0.0809			
50000	8.45	0.544	0.235	6.94	0.345	0.1492	5 825	0.226	0.0977			
60000	10.14	0.766	0.332	8.43	0.4915	0.2343	6.99	0.319	0.1382			
70000				9.82	0.655	0.2835	8.155	0.422	0.1828			
80000				11.21	0.8292	0.358	9.320	0.548	0.2356			
90000							10.485	0.682	0.295			
100000							11.650	0.83	0.359			

Friction in Elbows

Gallons per min.	Velocity in feet per sec.	Friction Head in feet	Friction loss in lbs. per sq. in.	Velocity in feet per sec.	Friction Head in feet	Friction loss in lbs. per sq. in.	Velocity in feet per sec.	Friction Head in feet	Friction loss in lbs. per sq. in.	Velocity in feet per sec.	Friction Head in feet.	Friction loss in lbs. per sq. in.
		¾″			1″			1¼″			1½″	
5	3.64	0.161	0.07	2.04	0.0624	0.027	1.3	0.0185	0.008			
10	7.28	0.644	.28	4.08	0.217	0.094	2.6	0.0716	0.031			
15	10.92	1.449	0.63	6.12	0.488	0.212	3.9	0.1592	0.069	2.73	0.092	0.04
20	14.56	2.576	1.12	8.16	0.8675	0.376	5.2	0.2835	0.123	3.64	0.1591	0.069
25	18.2	4.002	1.74	10.20	1.35	0.585	6.5	0.448	0.194	4.55	0.249	0.108
30				12.24	1.95	0.845	7.8	0.642	0.278	5.46	0.362	0.157
35		4″		14.28	2.65	1.15	9.1	0.877	0.380	6.37	0.495	0.215
40	1.04	0.01614	0.007	16.32	3.46	1.5	10.4	1.143	0.495	7.28	0.641	0.278
45	1.17	0.0208	0.009				11.7	1.445	0.626	8.19	0.81	0.352
50	1.3	0.0231	0.01		5″		13.	1.778	0.77	9.1	0.99	0.43
60	1.56	0.0345	0.015	0.88	0.01382	0.006	15.6	2.562	1.11	10.92	1.429	0.62
70	1.82	0.0485	0.021	1.04	0.0208	0.009	18.2	3.51	1.52	12.74	1.82	0.86
75	1.95	0.0552	0.024	1.2	0.0231	0.01	19.5	4.02	1.74	13.65	2.26	0.98
80	2.08	0.0623	0.027	1.28	0.0277	0.012	20.8	4.57	1.98	14.56	2.56	1.11
90	2.34	0.0808	0.035	1.44	0.0323	0.014		6″		16.38	3.25	1.41
100	2.60	0.0992	0.043	1.6	0.0392	0.017	1.14	0.01845	0.008			
125	3.25	0.1546	0.067	2.	0.0622	0.027	1.42	0.03	0.013		7″	
150	3.8	0.2215	0.096	2 4	0.09	0.039	1.71	0.0438	0.019	1.2	0.0231	0.01
175	4.45	0.3048	0.132	2.8	0.1222	0.053	2.	0.06	0.026	1.38	.0326	0.010
185	4.7	0.348	0.1509	2.96	0.1342	0.0582	2.11	0.068	0.0294	1.55	0.349	0.0149
200	5.1	0.397	0.172	3.2	0.157	0.068	2.28	0.0738	0.032	1.7	0.0462	0.02
250	6.4	0.6195	0.268	4.	0 252	0.109	2.8	0.11967	0.0518	2.1	0.0669	0.029
265	6.79	0.7025	0.34	4.24	0.2743	0.119	3.03	0.1404	0.067	2.23	0.076	0.0339
300	7.6	0.886	0.384	4 8	0.36	0.156	3.4	0.1754	0.076	2.4	0.0969	0.042
350	8.9	1.244	0.530	5.6	0 496	0.215	3.9	0.2375	0.103	2.94	0.1314	0.057
370	9.41	1.356	0.587	5.92	0.535	0.232	4.22	0.273	0.1182	3.1	0.149	0.0645
400	10.1	1.586	0.688	6.5	0.6275	0.272	4.5	0.295	0.128	3.36	0.1845	0.08
450	11.4	2.015	0.870	7.3	0.811	0.352	5.05	0.392	0.170	3.96	0.217	0.094
480	12.16	2.256	0.976	7.68	0.901	0.39	5.49	0.4602	0.1994	4.03	0.2454	0.1062
500	12.7	2.47	1.07	8.1	1.007	0.436	5.6	0.481	0.208	4.2	0.2675	0.116
750		14″		12.1	2.24	0.970	8.4	1.085	0.470	6.3	0.6	0.260
1000	2.224	0.0772	0.0335	16.1	4.02	1.74	11.3	1.92	0.832	8.4	1.069	0.464
1100	2.444	0.0917	0.039				12.57	2.413	1.045	9.25	1.329	0.575
1250	2.78	0.1185	0.0517		15″		14.1	3.025	1.31	10.5	1.676	0.728
1500	3.336	0.1695	0.0738	2.7	0.112	0.0486		18″		12.6	1.938	0.84
2000	4.448	0.303	0.131	3.6	0.1985	0.0859	2.74	0.115	0.0498			
2200	4.9	0.367	0.159	3.96	0.249	0.108	3.18	0.1548	0.066			
2500	5.562	0.475	0.206	4.5	0.311	0.1347	3.425	0.1795	0.0779		20″	
3000	6.674	0.690	0.3	5.4	0.4485	0.1944	4.11	0.258	0.112	3.308	0.168	0.073
3300	7.34	0.8215	0.353	5.94	0.539	0.233	4.52	0.313	0.1357	3.64	0.203	0.0883
3500	7.786	0.927	0.402	6.3	0.609	0.264	4.795	0.352	0.1525	3.86	3.228	0.0992
4000	8.898	1.206	0.526	7.2	0.795	0.3435	5.48	0.462	0.2	4.41	0.298	0.0998
4500	10.013	1.539	0.665	8.1	1.016	0.442	6.165	0.583	0.253	4.955	0.378	0.164
5000	12.237	2.29	0.997	9.	1.243	0.5395	6.85	0.719	0.312	5.51	0.4675	0.232
6000	14.461	3.22	1.39	10.8	1.79	0.774	8.22	1.03	0.447	6.61	0.671	0.292
7300				13.13	2.639	1.143	10.	1.555	0.674	8.05	0.984	0.4275
7500		30″		13.5	2.798	1.212	10.275	1.615	0.701	8.26	1.048	0.4556
10000	4.77	0.35	0.152				13.70	2.88	1.25	11.02	1.87	0.814
10500	5.10	0.3995	0.1735		36″		14.4	3.18	1.378	11.58	2.095	0.911
14000	6.69	0.686	0.298				19.	5.64	2.44	15.43	3.67	1.595
15000	7.16	0.778	0.338	4.93	0.374	0.162	20.55	6.49	2.81			
17000	8.11	1.246	1.591	5.58	0.4765	0.2063						
20000	9.55	1.40	0.609	6.57	0.662	0.287		40″				
25000	11.94	2.188	0.948	8.21	1.034	0.448	6.61	0.669	0.29		45″	
30000	14.32	3.15	1.37	9.95	1.515	0.656	7.93	0.963	0.417	6.255	0.6	0.26
35000				11.59	2.06	0.892	9.25	1.312	0.568	7.29	0.416	0.3525
40000				13.23	2.68	1.164	10.57	1.71	0.7395	8.33	1.031	0.46
45000				14.87	3.39	1.467	11.89	2 17	0.94	9.37	1.349	0 585
50000				16.51	4.19	1.815	13.21	2.68	1.161	10.41	1.669	0.721
60000				19.80	6.04	2.62	15.85	3.85	1.665	12.495	2.39	1.035
70000				23.09	8 19	3.55	18.49	5.25	2.275	14.58	3.25	1.41
80000				26.38	10.64	4.62	22.18	7.53	3.26	16.665	4.26	1.85
90000				29.67	13.5	5.85	24.77	9.44	4.09	18.75	5.39	2.34
100000				32.96	16.62	7.41	27.41	11.52	5.00	20.835	6.68	2.89

NOTE—Calculated from Weisback's formula for very short bends with a radius equal to the radius of the pipe.

Friction in Elbows

(Continued)

Velocity in feet per sec.	Friction Head in feet	Friction loss in lbs. per sq. in.	Velocity in feet per sec.	Friction Head in feet	Friction loss in lbs. per sq. in.	Velocity in feet per sec.	Friction Head in feet	Friction loss in lbs. per sq. in.	Velocity in feet per sec.	Friction Head in feet	Friction loss in lbs. per sq. in.	Gallons per min.
												5
												10
												15
	2″											20
2.6	0.087	0.038										25
3.03	0.127	0.055		2½″								30
3.54	0.1755	0.076	2.29	0.0854	0.037							35
4.05	0.2264	0.098	2.62	0.1131	0.049		3″					40
4.56	0.288	0.125	2.95	0.143	0.062	2.016	0.0599	0.026				45
5.1	0.353	0.153	3.3	0.1848	0.08	2.24	0.0738	0.032		3½″		50
6.12	0.5075	0.22	3.95	0.258	0.112	2.688	0.1015	0.044	1.992	0.06	0.026	60
7.14	0.701	0.304	4.6	0.3415	0.148	3.136	3.1383	0.06	2.324	0.0808	0.035	70
7.7	0.808	0.35	4.93	0.397	0.172	3.360	0.166	0.072	2.490	0.0923	0.04	75
8.16	0.925	0.392	5.26	0.452	0.196	3.584	0.1845	0.08	2.656	0.1014	0.044	80
9.18	1.155	0.5	5.91	0.573	0.248	4.032	0.2395	0.104	2.988	0.1385	0.06	90
10.2	1.412	0.612	6.5	0.738	0.32	4.480	0.295	0.128	3.320	0.1569	0.068	100
12.8	2.24	0.97	8.13	1.175	0.48	5.6	0.462	0.2	4.15	0.258	0.112	125
15.3	3.205	1.39	9.8	1.58	0.685	6.8	0.659	0.286	4.98	0.3695	0.16	150
			11.43	2.16	0.935	7.92	0.899	0.390	5.81	0.504	0.218	175
			11.84	2.42	1.049	8.34	1.055	0.457	6.14	0.5745	0.248	185
	8″		13.06	2.955	1.28	9.04	1.18	0.512	6.64	0.628	0.272	200
1.6	0.0392	0.017				11.28	1.845	0.8	8.30	1.029	0.446	250
1.698	0.0441	0.191		9″		12.4	2.35	1.018	8.8	1.183	0.513	265
1.9	0.0577	0.025	1.5	0.0369	0.016	13.52	2.63	1.14	9.96	1.478	0.64	300
2.2	0.0784	0.034	1.75	0.0507	0.022				11.62	2.03	0.88	350
2.37	0.0843	0.365	1.85	0.0573	0.0248				12.28	2.302	0.997	370
2.6	0.1015	0.044	2.	0.0646	0.028		10″		13.28	2.515	1.09	400
2.92	0.1315	0.057	2.25	0.083	0.036	1.8	0.053	0.023				450
3.07	0.144	0.0624	2.4	0.088	0.038	1.92	0.0563	0.0244		12″		480
3.2	0.1568	0.068	2.5	0.1015	0.044	2.	0.065	0.028	1.4	0.0369	0.016	500
4.8	0.3598	0.156	3.75	0.231	0.102	3.	0.1453	0.063	2.1	0.0715	0.031	750
6.4	0.627	0.272	5.	0.406	0.176	4.	0.258	0.112	2.8	0.1439	0.064	1000
7.04	0.756	0.3295	5.5	0.4615	0.2	4.4	0.293	0.127	3.08	0.145	0.6028	1100
8.0	1.005	0.435	6.25	0.637	0.276	5.	0.403	0.175	3.5	0.1981	0.086	1250
9.6	1.44	0.624	7.5	0.923	0.4	6.1	0.58	0.252	4.2	0.286	0 124	1500
12.7	2.44	1.055	10.	1.535	0.665	8.1	1.008	0.436	5.6	0.4747	0.205	2000
14.09	3.025	1.308	11.	1.85	0.815	8.8	1.183	0.513	6.16	0.5805	0.2518	2200
15.9	3.88	1.68	12.5	2.395	1.035	10.1	1.568	0.68	7.	0.752	0.327	2500
			15.	3.46	1.5	12.1	2.248	0.975	8.4	1.082	0.429	3000
	22″					13.2	2.63	1.14	9.24	1.34	0.5802	3300
3.15	0.1525	0.0664		24″		14.1	3.05	1.325	9.8	1.471	0.638	3500
3.6	0.1992	0.0867	3.08	0.146	0.0685				1.92	1.92	0.833	4000
4.06	0.253	1.10	3.47	0.1873	0.0815				12.6	2.435	1.055	4500
4.5	0 312	0 1344	3.855	0.2285	0.0994				14.	3.01	1.34	5000
5.4	0.0448	0.194	4.62	0.3288	0.143							6000
6.57	0.66	0.286	5.625	0.486	0.2112							7310
6.75	0.698	0.303	5.783	0.564	0.245		26″					7500
9.	1.241	0.539	7.708	0.912	0.396	6.38	.625	0.271				10000
9.45	1.36	0.589	8.1	1.01	0.4385	6.7	0.6866	0.297				10500
12.6	2.578	1.117	10.79	1.788	0.777	8.93	1.223	0.53		28″		14000
13.5	2.8	1.212	11.56	2.05	0.891	9.57	1.408	0.609	8.25	1.09	0.472	15000
15.3	3.568	1.546	13.10	2.64	1.147	10.84	1.95	0.845	9.35	1.334	0.578	17000
18.	4.96	2.15	15.4	3.64	1.58	12.76	2.49	1.084	11.00	1.855	0.806	20000
22.5	7.76	3.355				17.05	4.47	1.939	13.75	2.945	1.208	25000
						20.24	6.18	2.68	16.50	4.16	1.802	30000
	50″			55″		25.52	9.98	4.32	19.25	5.68	2.46	35000
6.76	0.680	0.295	5.56	0.475	0.206				22.00	7.43	3.225	40000
7.605	0.890	0.385	6.25	0.599	0.259		60″		24.75	9.89	4.07	45000
8.45	1.095	0.475	6.94	0.730	0.316	5.825	0.52	0.226	27.50	11.6	5.2	50000
10.14	1.157	0.680	8.43	1.09	0.473	6.790	0.749	0.324				60000
11.83	2.14	0.930	9.82	1.48	0.642	8.155	1.02	0.4425				70000
13.52	2.81	1.215	11.21	1.93	0.833	9.320	1.33	0.575				80000
15.21	3.56	1.54	12.60	2.43	1.055	10.485	1.7	0.735				90000
16.90	4.4	1.91	13.99	3.00	1.300	11.650	2.08	0.91				100000

NOTE—Calculated from Weisback's formula for very short bends with a radius equal to the radius of the pipe.

Friction of Standard Pipe Fittings

The following equivalent length of straight pipe snould be added for each fitting in figuring friction:

Size of Fittings	¾	1	1¼	1½	2	2½	3	4	5	6
Elbows	5	5	6	7	7	10	12	18	25	30
Return Bends	10	10	12	14	14	20	24	36	50	60
Globe Valves	6	6	7	8	8	12	24	30	40	50

Owing to the burr (caused by cutting the pipe with wneel cutter) obstructing the flow in the smaller pipes, it is advisable, unless the burrs are reamed out, to multiply the above figures, by 3 for ¾ and 1 inch fittings and by 2 for 1¼, 1½ and 2 inch fittings.

Equation of Pipes

The following table gives the actual dimcnsions of standard "Merchant" wrought iron pipe with the inside diameter in inches of each branch in a series of equal branches, whose total internal cross sectional area is equal to that of the main pipe. Thus if it is desired to find the size of pipe required for four branches, whose internal cross sectional area is equal to that of a 3 inch pipe, by referring to the table opposite the 3 inch pipe and under 4, the diameter 1.533 inches is found, which is the required dimension. By reference to the column of diameters, it will be seen that the proper size of pipes will be 1½ inches, the diameter of this size being 1.611 inches. No account of friction is taken in this table, which must be considered in any actual problem.

Size Pipe	Actual Internal Area		Actual Inside Diam.	Diameter in inches of each branch of the following number of equivalent branches									Size Pipe
	Square Inches	Square Feet	Inches	2	3	4	5	6	7	8	9	10	
⅛	.0568	.0004	.27	.....	.....	.....	.....	.....	.....	.....	.....	.....	⅛
¼	.1041	.0007	.364	.257	.210	.....	.....	.....	.....	.....	.....	.....	¼
⅜	.1909	.0013	.484	.349	.285	.247	.220	.....	.....	.....	.....	.....	⅜
½	.3039	.0021	.623	.440	.359	.311	.279	.254	.231	.220	.207	.....	½
¾	.5333	.0037	.824	.582	.475	.412	.368	.336	.311	.291	.274	.260	¾
1	.8609	.0060	1.048	.741	.605	.524	.468	.427	.400	.375	.349	.331	1
1¼	1.4957	.0104	1.380	.976	.796	.690	.617	.563	.521	.488	.460	.436	1¼
1½	2.036	.0141	1.611	1.139	.930	.805	.720	.657	.608	.570	.537	.594	1½
2	3.356	.0233	2.067	1.461	1.193	1.033	.924	.843	.781	.730	.689	.653	2
2½	4.780	.0332	2.468	1.745	1.424	1.234	1.103	1.007	.932	.872	.822	.780	2½
3	7.383	.0513	3.067	2.169	1.770	1.533	1.372	1.252	1.159	1.084	1.022	.970	3
3½	9.886	.0687	3.548	2.509	2.048	1.774	1.586	1.448	1.340	1.253	1.182	1.123	3½
4	12.730	.0884	4.026	2.847	2.330	2.013	1.809	1.643	1.521	1.423	1.342	1.273	4
4½	15.961	.1108	4.508	3.185	2.602	2.254	2.016	1.840	1.703	1.594	1.502	1.425	4½
5	19.986	.1388	5.045	3.568	2.912	2.522	2.256	2 059	1.906	1.784	1.681	1.595	5
6	28.886	.2006	6.065	4.289	3.501	3.032	2.716	2.475	2.292	2.144	2.021	1.918	6
7	38.743	.2690	7.023	4.966	4.054	3.511	3.140	2.867	2.654	2.483	2.341	2.228	7
8	50.021	.3474	7.982	5.645	4.618	3.991	3.570	3.258	3.013	2.822	2.661	2.524	8
10	78.822	.5474	10.019	7.085	5.785	5.009	4.462	4.090	3.783	3.603	3.339	3.168	10
12	113.088	.7854	12.000	8.486	6.928	6.000	5.367	4.898	4.535	4.243	4.000	3.794	12
14	159.485	1.1075	13.250	9.370	7.650	6.625	5.926	5.409	5.007	4.684	4.410	4.189	14

Theoretical Discharge of Nozzles in U. S. Gallons per Minute

Head		Veloc'y of Disch. Ft. per Sec.	DIAMETER OF NOZZLE IN INCHES												
Lbs	Feet		1/16	1/8	3/16	1/4	3/8	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8
10	23.1	38.6	0.37	1.48	3.32	5.91	13.3	23.6	36.9	53.1	72.4	94.5	120	148	179
15	34.6	47.25	0.45	1.81	4.06	7.24	16.3	28.9	45.2	65.0	88.5	116.	147	181	219
20	46.2	54.55	0.52	2.09	4.69	8.35	18.8	33.4	52.2	75.1	102.	134.	169	209	253
25	57.7	61.0	0.58	2.34	5.25	9.34	21.0	37.3	58.3	84.0	114.	149.	189	234	283
30	69.3	66.85	0.64	2.56	5.75	10.2	23.0	40.9	63.9	92.0	125.	164.	207	256	309
35	80.8	72.2	0.69	2.77	6.21	11.1	24.8	44.2	69.0	99.5	135.	177.	224	277	334
40	92.4	77.2	0.74	2.96	6.64	11.8	26.6	47.3	73.8	106.	145.	189.	239	296	357
45	103.9	81.8	0.78	3.13	7.03	12.5	28.2	50.1	78.2	113.	153.	200.	253	313	379
50	115.5	86.25	0.83	3.30	7.41	13.2	29.7	52.8	82.5	119.	162.	211.	267	330	399
55	127.0	90.4	0.87	3.46	7.77	13.8	31.1	55.3	86.4	125.	169.	221.	280	346	418
60	138.6	94.5	0.90	3.62	8.12	14.5	32.5	57.8	90.4	130.	177.	231.	293	362	438
65	150.1	98.3	0.94	3.77	8.45	15.1	33.8	60.2	94.0	136.	184.	241.	305	376	455
70	161.7	102.1	0.98	3.91	8.78	15.7	35.2	62.5	97.7	141.	191.	250.	317	391	473
75	173.2	105.7	1.01	4.05	9.08	16.2	36.4	64.7	101.	146.	198.	259.	327	404	489
80	184.8	109.1	1.05	4.18	9.39	16.7	37.6	66.8	104.	150.	205.	267.	338	418	505
85	196.3	112.5	1.08	4.31	9.67	17.3	38.8	68.9	108.	155.	211.	276.	349	431	521
90	207.9	115.8	1.11	4.43	9.95	17.7	39.9	70.8	111.	160.	217.	284.	359	443	536
95	219.4	119.0	1.14	4.56	10.2	18.2	41.0	72.8	114.	164.	223.	292.	369	456	551
100	230.9	122.0	1.17	4.67	10.5	18.7	42.1	74.7	117.	168.	229.	299.	378	467	565
105	242.4	125.0	1.20	4.79	10.8	19.2	43.1	76.5	120.	172.	234.	306.	388	479	579
110	254.0	128.0	1.23	4.90	11.0	19.6	44.1	78.4	122.	176.	240.	314.	397	490	593
115	265.5	130.9	1.25	5.01	11.2	20.0	45.1	80.1	125.	180.	245.	320.	406	501	606
120	277.1	133.7	1.28	5.12	11.5	20.5	46.0	81.8	128.	184.	251.	327.	414	512	619
125	288.6	136.4	1.31	5.22	11.7	20.9	47.0	83.5	130.	188.	256.	334.	423	522	632
130	300.2	139.1	1.33	5.33	12.0	21.3	48.0	85.2	133.	192.	261.	341.	432	533	645
135	311.7	141.8	1.36	5.43	12.2	21.7	48.9	86.7	136.	195.	266.	347.	439	543	656
140	323.3	144.3	1.38	5.53	12.4	22.1	49.8	88.4	138.	199.	271.	354.	448	553	668
145	334.8	146.9	1.41	5.62	12.6	22.5	50.6	89.9	140.	202.	275.	360.	455	562	680
150	346.4	149.5	1.43	5.72	12.9	22.9	51.5	91.5	143.	206.	280.	366.	463	572	692
175	404.1	161.4	1.55	6.18	13.9	24.7	55.6	98.8	154.	222.	302.	395.	500	618	747
200	461.9	172.6	1.65	6.61	14.8	26.4	59.5	106.	165.	238.	323.	423.	535	660	799
250	577.4	193.0	1.85	7.39	16.6	29.6	66.5	118.	185.	266.	362.	473.	598	739	894
300	692.8	211.2	2.02	8.08	18 2	32.4	72.8	129.	202.	291.	396.	517.	655	808	977

Head		Veloc'y of Disch. Ft. per Sec.	DIAMETER OF NOZZLE IN INCHES												
Lbs	Feet		1 1/2	1 3/4	2	2 1/4	2 1/2	2 3/4	3	3 1/2	4	4 1/2	5	5 1/2	6
10	23.1	38.6	213	289	378	479	591	714	851	1158	1510	1915	2365	2855	3405
15	34.6	47.25	260	354	463	585	723	874	1041	1418	1850	2345	2890	3490	4165
20	46.2	54.55	301	409	535	676	835	1009	1203	1638	2135	2710	3340	4040	4810
25	57.7	61.0	336	458	598	756	934	1128	1345	1830	2385	3025	3730	4510	5380
30	69.3	66.85	368	501	655	828	1023	1236	1473	2005	2615	3315	4090	4940	5895
35	80.8	72.2	398	541	708	895	1106	1335	1591	2168	2825	3580	4415	5340	6370
40	92.4	77.2	425	578	756	957	1182	1428	1701	2315	3020	3830	4725	5710	6810
45	103.9	81.8	451	613	801	1015	1252	1512	1802	2455	3200	4055	5000	6050	7210
50	115.5	86.25	475	647	845	1070	1320	1595	1900	2590	3375	4275	5280	6380	7600
55	127.0	90.4	498	678	886	1121	1385	1671	1991	2710	3540	4480	5530	6690	7970
60	138.6	94.5	521	708	926	1172	1447	1748	2085	2835	3700	4685	5790	6980	8330
65	150.1	98.3	542	737	964	1220	1506	1819	2165	2950	3850	4875	6020	7270	8670
70	161.7	102.1	563	765	1001	1267	1565	1888	2250	3065	4000	5060	6250	7560	9000
75	173.2	105.7	582	792	1037	1310	1619	1955	2330	3170	4135	5240	6475	7820	9320
80	184.8	109.1	602	818	1070	1354	1672	2020	2405	3280	4270	5410	6690	8080	9630
85	196.3	112.5	620	844	1103	1395	1723	2080	2480	3375	4400	5575	6890	8320	9920
90	207.9	115.8	638	868	1136	1436	1773	2140	2550	3475	4530	5740	7090	8560	10210
95	219.4	119.0	656	892	1168	1476	1824	2200	2625	3570	4655	5900	7290	8800	10500
100	230.9	122.0	672	915	1196	1512	1870	2255	2690	3660	4775	6050	7470	9030	10770
105	242.4	125.0	689	937	1226	1550	1916	2312	2755	3750	4890	6200	7650	9250	11020
110	254.0	128.0	705	960	1255	1588	1961	2366	2820	3840	5010	6350	7840	9470	11300
115	265.5	130.9	720	980	1282	1621	2005	2420	2885	3930	5120	6490	8010	9680	11550
120	277.1	133.7	736	1002	1310	1659	2050	2470	2945	4015	5225	6630	8180	9900	11800
125	288.6	136.4	751	1022	1338	1690	2090	2520	3005	4090	5340	6760	8350	10100	12030
130	300.2	139.1	767	1043	1365	1726	2132	2575	3070	4175	5450	6900	8530	10300	12290
135	311.7	141.8	780	1063	1390	1759	2173	2620	3125	4250	5550	7030	8680	10490	12510
140	323.3	144.3	795	1082	1415	1790	2212	2670	3180	4330	5650	7160	8850	10690	12730
145	334.8	146.9	809	1100	1440	1820	2250	2715	3235	4410	5740	7280	8990	10880	12960
150	346.4	149.5	824	1120	1466	1853	2290	2760	3295	4485	5850	7410	9150	11070	13200
175	404.1	161.4	890	1210	1582	2000	2473	2985	3560	4840	6310	8000	9890	11940	14250
200	461.9	172.6	950	1294	1691	2140	2645	3190	3800	5175	6750	8550	10580	12770	15220
250	577.4	193.0	1063	1447	1891	2392	2955	3570	4250	5795	7550	9570	11820	14290	17020
300	692.8	211.2	1163	1582	2070	2615	3235	3900	4650	6330	8260	10480	12940	15620	18610

NOTE—The actual quantities will vary from these figures, the amount of variation depending upon the shape of nozzle and size of pipe at the point where the pressure is determined.

Hydrant and Hose Stream Data

From Tables Published by John R. Freeman, M.E.

Pressure at Nozzle	Gallons Discharged per Min.	Vertical Distance of Stream	Horizontal Distance of Stream	Pressure in pounds required at Hydrant or Pump to maintain pressure at nozzle through various lengths of 2½ inch smooth, rubber-lined hose.								
				50 ft.	100 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	800 ft.	1000 ft.
¾ INCH SMOOTH NOZZLE												
35	97	55	41	37	38	40	42	44	46	48	53	57
40	104	60	44	42	43	46	48	50	53	55	60	65
45	110	64	47	47	48	51	54	57	59	62	68	73
50	116	67	50	52	54	57	60	63	66	69	75	81
55	122	70	52	58	59	63	66	69	73	76	83	89
60	127	72	54	63	65	68	72	76	79	83	90	97
65	132	74	56	68	70	74	78	82	86	90	98	106
70	137	76	58	73	75	80	84	88	92	97	105	114
75	142	78	60	79	81	85	90	94	99	104	113	122
80	147	79	62	84	86	91	96	101	106	111	120	130
85	151	80	64	89	92	97	102	107	112	117	128	138
90	156	81	65	94	97	102	108	113	119	124	135	146
95	160	82	66	99	102	108	114	120	125	131	143	154
100	164	83	68	105	108	114	120	126	132	138	150	163
⅞ INCH SMOOTH NOZZLE												
35	133	56	46	38	40	44	48	52	56	60	68	76
40	142	62	49	43	46	50	55	59	64	68	78	87
45	150	67	52	49	51	57	62	67	72	77	87	97
50	159	71	55	54	57	63	69	74	80	86	97	108
55	166	74	58	60	63	69	75	82	88	94	107	119
60	174	77	61	65	69	75	82	89	96	103	116	130
65	181	79	64	71	74	82	89	96	104	111	126	141
70	188	81	66	76	80	88	96	104	112	120	136	152
75	194	83	68	82	86	94	103	111	120	128	145	162
80	201	85	70	87	91	101	110	119	128	137	155	173
85	207	87	72	92	97	107	116	126	136	145	165	184
90	213	88	74	98	103	113	123	134	144	154	174	195
95	219	89	75	103	109	119	130	141	152	163	184	206
100	224	90	76	109	114	126	137	148	160	171	194	216
1 INCH SMOOTH NOZZLE												
35	174	58	51	40	44	51	57	64	71	78	92	105
40	186	64	55	46	50	58	66	73	81	89	105	120
45	198	69	58	52	56	65	74	83	91	100	118	135
50	208	73	61	57	62	72	82	92	102	111	131	151
55	218	76	64	63	69	79	90	101	112	122	144	166
60	228	79	67	69	75	87	98	110	122	134	157	181
65	237	82	70	75	81	94	107	119	132	145	170	196
70	246	85	72	80	87	101	115	128	142	156	183	211
75	255	87	74	86	94	108	123	138	152	167	196	226
80	263	89	76	92	100	115	131	147	162	178	209	241
85	274	91	78	98	106	123	139	156	173	189	222	...
90	279	92	80	103	112	130	147	165	183	200	236	...
95	287	94	82	109	118	137	156	174	193	211	249	...
100	295	96	83	115	125	144	164	183	203	223	...	...

The pressures given are indicated pressures, not effective pressures. Effective pressures would be slightly greater.

Hydrant and Hose Stream Data

(Continued)

From Tables Published by John R. Freeman, M. E.

Pressure at Nozzle	Gallons Discharged per Min.	Vertical Discharge of Stream	Horizontal Discharge of Stream	Pressure in pounds required at Hydrant or Pump to maintain pressure at nozzle through various lengths of 2½ inch smooth rubber-lined hose.								
				50 ft.	100 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	800 ft.	1000 ft.
1⅛ INCH SMOOTH NOZZLE												
35	222	59	54	43	49	60	71	82	94	105	127	149
40	238	65	59	50	56	69	81	94	07	120	145	171
45	252	70	63	56	63	77	92	106	120	135	163	192
50	266	75	66	62	70	86	102	118	134	150	181	213
55	279	80	69	68	77	95	112	130	147	165	200	235
60	291	83	72	74	84	103	122	141	160	180	218	256
65	303	86	75	81	91	112	132	153	174	195	236	...
70	314	88	77	87	98	120	143	165	187	209	254	...
75	325	90	79	93	105	129	153	177	201	224	...	...
80	336	92	81	99	112	138	163	188	214	239	...	...
85	346	94	83	106	119	146	173	200	227	254	...	...
90	356	96	85	112	126	155	183	212	241	...	...	...
95	366	98	87	118	133	163	194	224	254	...	...	...
100	376	99	89	124	140	172	204	236	...	...	...	...
1¼ INCH SMOOTH NOZZLE												
35	277	60	59	48	57	74	91	109	126	142	178	212
40	296	67	63	55	65	84	104	124	144	164	203	243
45	314	72	67	62	73	95	117	140	162	184	229	...
50	331	77	70	68	81	106	130	155	180	204	254	...
55	347	81	73	75	89	116	143	170	198	225	...	...
60	363	85	76	82	97	127	156	186	216	245	...	...
65	377	88	79	89	105	137	169	201	234	...	...	...
70	392	91	81	96	113	148	182	217	252	...	...	...
75	405	93	83	103	121	158	195	232	...	...	...	...
80	419	95	85	110	129	169	208	148	...	...	...	...
85	432	97	88	116	137	179	221	...	...	...	...	...
90	444	99	90	123	145	190	234	...	...	...	...	...
95	456	100	92	130	154	201	247	...	...	...	...	...
100	468	101	93	137	162	211	261	...	...	...	...	...
1⅜ INCH SMOOTH NOZZLE												
35	340	62	62	54	67	94	120	146	172	198	250	...
40	363	69	66	62	77	107	137	166	196	226	...	...
45	385	74	70	70	87	120	154	187	221	254	...	...
50	406	79	73	78	96	134	171	208	245	...	...	...
55	426	83	76	86	106	147	188	229	270	...	...	...
60	445	87	79	93	116	160	205	250	...	...	...	...
65	463	90	82	101	125	174	222	...	...	...	...	...
70	480	92	84	109	135	187	239	...	...	...	...	...
75	497	95	86	117	145	201	256	...	...	...	...	...
80	514	97	88	124	154	214	...	...	...	...	...	...
85	529	99	90	132	164	227	...	...	...	...	...	...
90	545	100	92	140	173	240	...	...	...	...	...	...
95	560	101	94	148	183	254	...	...	...	...	...	...
100	574	103	96	156	193	...	...	...	...	...	...	...

The pressures given are indicated pressures, not effective pressures. Effective pressures would be slightly greater.

Table Converting Inches Vacuum, into Feet Suction

Inch Vac.	Feet	Inch Vac.	Feet	Inch Vac.	Feet	Inch Vac.	Feet
¼	0.28	8¼	9.35	16¼	18.42	24¼	27.50
½	0.56	8½	9.64	½	18.71	½	27.78
¾	0.85	8¾	9.92	¾	18.99	¾	28.07
1	1.13	9	10.21	17	19.28	25	28.35
1¼	1.41	¼	10.49	¼	19.56	¼	28.63
1½	1.70	½	10.77	½	19.84	½	28.91
1¾	1.98	¾	11.06	¾	20.13	¾	29.20
2	2.27	10	11.34	18	20.41	26	29.48
2¼	2.55	¼	11.62	¼	20.70	¼	29.76
2½	2.84	½	11.90	½	20.98	½	30.05
2¼	3.12	¾	12.19	¾	21.27	¾	30.33
3	3.41	11	12.47	19	21.55	27	30.62
3¼	3.69	¼	12.75	¼	21.83	¼	30.90
3½	3.98	½	13.04	½	22.11	½	31.19
3¾	4.26	¾	13.32	¾	22.40	¾	31.47
4	4.54	12	13.61	20	22.68	28	31.75
4¼	4.82	¼	13.89	¼	22.96	¼	32.03
4½	5.11	½	14.18	½	23.24	½	32.32
4¾	5.39	¾	14.46	¾	23.53	¾	32.60
5	5.67	13	14.74	21	23.81	29	32.89
5¼	5.95	¼	15.02	¼	24.09	¼	33.17
5½	6.23	½	15.31	½	24.38	½	33.46
5¾	6.52	¾	15.59	¾	24.66	¾	33.74
6	6.80	14	15.88	22	24.95	30	.....
6¼	7.08	¼	16.16	¼	25.23	....	....
6½	7.37	½	16.45	½	25.51	.....	.....
6¾	7.65	¾	16.73	¾	25.80	.....	.....
7	7.94	15	17.01	23	26.08	.....	.....
7¼	8.22	¼	17.29	¼	26.36	.....	.....
7½	8.50	½	17.57	½	26.65	.....	.....
7¾	8.79	¾	17.86	¾	26.93	.....	.....
8	9.07	16	18.14	24	27.22	.....	.....

To convert inches vacuum into feet, multiply by 1.13.

Relative Quantities of Water

Delivered in 24 hours, in 1 hour and in 1 minute.

Gal's in 24 hours	Gal's in 1 hour	Gal's in 1 min.	Gal's in 24 hours	Gal's in 1 hour	Gal's in 1 min.	Gal's in 24 hours	Gal's in 1 hour	Gal's in 1 min
2500000	104166.0	1736.0	650000	27083.3	451.3	150000	6250 0	104 1
2000000	83333.3	1388.0	600000	25000.0	416.7	100000	4166.6	69 4
1500000	62500.0	1041.7	550000	22916.6	381.9	75000	3125.0	52 9
1000000	41666.0	694.3	500000	20833.3	347.2	60000	2500.0	41 6
950000	39583.3	659.7	450000	18750.0	312.5	50000	2083.0	34.7
900000	37500 0	625.0	400000	16666.6	277.7	25000	1041.6	17.3
850000	35416.6	590.2	350000	14583.3	243.0	20000	833.3	13.8
800000	33333.3	555.5	300000	12500.0	208.3	15000	625.0	10.4
750000	31250.0	520.8	250000	10416.7	173.6	10000	416.6	6.9
700000	29166 6	486.1	200000	8333 0	138.8	5000	208.3	3 4

Rules for Determining Size and Speed of Pulleys or Gears

The driving pulley is called the Driver, and the driven pulley the Driven.

If the number of teeth in gears are used instead of diameter, in these calculations, number of teeth must be substituted wherever diameter occurs.

To determine the diameter of Driver, the diameter of the Driven and its revolutions, and also revolutions of Driver being given.

$$\frac{\text{Diam. of Driven} \times \text{revolutions of Driven}}{\text{Revolutions of Driver.}} = \text{Diam. of Driver.}$$

To determine the diameter of Driven, the revolutions of the Driven and diameter and revolutions of the Driver being given.

$$\frac{\text{Diam. of Driver} \times \text{revolutions of Driver}}{\text{Revolutions of Driven.}} = \text{Diam. of Driven.}$$

To determine the revolutions of the Driver, the diameter and revolutions of the driven, and diameter of the Driver being given.

$$\frac{\text{Diam. of Driven} \times \text{revolutions of Driven}}{\text{Diameter of Driver.}} = \text{Rev. of Driver.}$$

To determine the revolutions of the Driven, the diameter and revolutions of the Driver, and diameter of the Driven being given

$$\frac{\text{Diam. of Driver} \times \text{revolutions of Driver}}{\text{Diameter of Driven.}} = \text{Rev. of Driven.}$$

Diameter of Pulleys with Corresponding Belt Speeds and Horse Power Belting will Transmit

Diameter of Pulley in Inches	Belt Speed in Feet per Minute per 100 R. P. M.	Corresponding Horse Power, Transmitted per 1 in. Belt Width	
		Single Belt	Double Belt
3.	78.6	.095	.190
3.82	100.	.121	.242
4.	105.	.127	.254
5.	131.	.158	.316
6.	157.	.190	.380
8.	210.	.254	.508
10.	262.	.317	.634
12.	314.	.380	.760
14.	366.	.443	.886
15.	393.	.475	.950
16.	419.	.507	1.01
18.	471.	.570	1.14
20.	524.	.634	1.27
22.	576.	.697	1.39
24.	628.	.760	1.52
26.	680.	.823	1.64
28.	733.	.888	1.77
30.	785.	.950	1.90
32.	838.	1.01	2.02
34.	890.	1.08	2.16
36.	942.	1.14	2.28
38.	995.	1.20	2.40
40.	1048.	1.26	2.52
42.	1100.	1.33	2.66
48.	1256.	1.52	3.04
54.	1415.	1.71	3.42
60.	1570.	1.90	3.80
66.	1722.	2.08	4.16
72.	1884.	2.28	4.56

R. P. M.=Revolutions per Minute.

To find the Belt Speed in Feet per Minute for any size pulley and any number of revolutions per minute: Multiply the diameter of the pulley in inches by the revolutions per minute, and multiply the product by .262.

To find the Horse Power for any width belt and any speed: Multiply the belt speed in feet per minute by the width of the belt in inches, and multiply the product by .00121 for a single ply belt, or by .00242 for a double ply belt. The final result is the horse power which the belt will transmit.

EXAMPLE: What is the speed of a belt running over a 42 inch pulley turning at 180 R. P. M.?

ANSWER: Belt speed $=42 \times 180 \times .262 \times 1980$ ft. per min.

EXAMPLE: What horse power will a 6 inch single ply belt transmit when traveling at this speed?

ANSWER: Horse power $=1980 \times 6 \times .00121 = 14.38$.

Note:—The horse power which a belt will transmit as given in the above tables is based on the assumption that the pulleys are both of equal diameter or nearly so. When one pulley is several times larger than the other, an extra allowance of width should be made to insure satisfactory transmission of the full amount of power. It is always best to have a liberal sized belt for all cases, as the liability of trouble of all kinds is minimized and the life of the belt is greatly prolonged.

Theoretical Horse Power Required to Raise Water to Different Heights

Feet Elevation / Gallons per Min.	5	10	15	20	25	30	35	40	45	50	60
5	.006	.012	.019	.025	.031	.037	.044	.05	.06	.06	.07
10	.012	.025	.037	.050	.062	.075	.087	.10	.11	.12	.15
15	.019	.037	.056	.075	.094	.112	.131	.15	.17	.19	.22
20	.025	.050	.075	.100	.125	.150	.175	.20	.22	.25	.30
25	.031	.062	.093	.125	.156	.187	.219	.25	.28	.31	.37
30	.037	.075	.112	.150	.187	.225	.262	.30	.34	.37	.45
35	.043	.087	.131	.175	.219	.262	.306	.35	.39	.44	.52
40	.050	.100	.150	.200	.250	.300	.350	.40	.45	.50	.60
45	.056	.112	.168	.225	.281	.337	.394	.45	.51	.56	.67
50	.062	.125	.187	.250	.312	.375	.437	.50	.56	.62	.75
60	.075	.150	.225	.300	.375	.450	.525	.60	.67	.75	.90
75	.093	.187	.281	.375	.469	.562	.656	.75	.84	.94	1.12
90	.112	.225	.337	.450	.562	.675	.787	.90	1.01	1.12	1.35
100	.125	.250	.375	.500	.625	.750	.875	1.00	1.12	1.25	1.50
125	.156	.312	.469	.625	.781	.937	1.094	1.25	1.41	1.56	1.87
150	.187	.375	.562	.750	.937	1.125	1.312	1.50	1.69	1.87	2.25
175	.219	.437	.656	.875	1.093	1.312	1.531	1.75	1.97	2.19	2.62
200	.250	.500	.750	1.000	1.250	1.500	1.750	2.00	2.25	2.50	3.00
250	.312	.625	.937	1.250	1.562	1.875	2.187	2.50	2.81	3.12	3.75
300	.375	.750	1.125	1.500	1.875	2.250	2.625	3.00	3.37	3.75	4.50
350	.437	.875	1.312	1.750	2.187	2.625	3.062	3.50	3.94	4.37	5.25
400	.500	1.000	1.500	2.000	2.500	3.000	3.500	4.00	4.50	5.00	6.00
500	.625	1.250	1.875	2.500	3.125	3.750	4.375	5.00	5.62	6.25	7.50

Feet Elevation / Gallons per Min.	75	90	100	125	150	175	200	250	300	350	400
5	.09	.11	.12	.16	.19	.22	.25	.31	.37	.44	.50
10	.19	.22	.25	.31	.37	.44	.50	.62	.75	.87	1.00
15	.28	.34	.37	.47	.56	.66	.75	.94	1.12	1.31	1.50
20	.37	.45	.50	.62	.75	.87	1.00	1.25	1.50	1.75	2.00
25	.47	.56	.62	.78	.94	1.09	1.25	1.56	1.87	2.19	2.50
30	.56	.67	.75	.94	1.12	1.31	1.50	1.87	2.25	2.62	3.00
35	.66	.79	.87	1.08	1.31	1.53	1.75	2.19	2.62	3.06	3.50
40	.75	.90	1.00	1.25	1.50	1.75	2.00	2.50	3.00	3.50	4.00
45	.84	1.01	1 12	1.41	1.69	1.97	2.25	2.81	3.37	3.94	4.50
50	.94	1.12	1.25	1.56	1.87	2.19	2.50	3.12	3.75	4.37	5.00
60	1.12	1.35	1.50	1.87	2.25	2.62	3.00	3.75	4.50	5.25	6.00
75	1.40	1.69	1.87	2.34	2.81	3.28	3.75	4.69	5.62	6.56	7.50
90	1.68	2.02	2.25	2.81	3.37	3.94	4.50	5.62	6.75	7.87	9.00
100	1.87	2.25	2.50	3.12	3.75	4.37	5.00	6.25	7.50	8.75	10.00
125	2.34	2.81	3.12	3.91	4.69	5.47	6.25	7.81	9.37	10.94	12.50
150	2.81	3.37	3.75	4.69	5.62	6.56	7.50	9.37	11.25	13.12	15.00
175	3.28	3.94	4.37	5.47	6.56	7.66	8.75	10.94	13.12	15.31	17.50
200	3.75	4.50	5.00	6.25	7.50	8.75	10.00	12.50	15.00	17.50	20.00
250	4.69	5.62	6.25	7.81	9.37	10.94	12.50	15.72	18.75	21.87	25.00
300	5.62	6.75	7.50	9.37	11.25	13.12	15.00	18.75	22.50	26.25	30.00
350	6.56	7.87	8.75	10.94	13.12	15.31	17.50	21.87	26.25	30.62	35.00
400	7.50	9.00	10.00	12.50	15.00	17.50	20.00	25.00	30.00	35.00	40.00
500	9.37	11.25	12.50	15.62	18.75	21.87	25.00	31.25	37.50	43.75	50.00

The theoretical horse power required to elevate water is found by multiplying the gallons pumped per minute by the total lift (including friction) in feet, and dividing by 4000.

Convenient Equivalents

1 second-foot equals 40 California miner's inches. (Law of March 23, 1901.)
1 second-foot equals 38.4 Colorado miner's inches.
1 second-foot equals 40 Arizona miner's inches.
1 second-foot equals 7.48 United States gallons per second; equals 448.8 gallons per minute, equals 646,272 gallons per day.
1 second-foot equals 6.23 British imperial gallons per second.
1 second-foot for one year covers one square mile 1.131 feet deep; 13.57 inches deep.
1 second-foot for one year equals 31,536,000 cubic feet.
1 second-foot equals about one acre-inch per hour.
1 second-foot falling 10 feet equals 1.136 horsepower.
100 California miner's inches equal 15.7 United States gallons per second.
100 California miner's inches equal 96.0 Colorado miner's inches.
100 California miner's inches for one day equal 4.96 acre-feet.
100 Colorado miner's inches equal 2.60 second-feet.
100 Colorado miner's inches equal 19.5 United States gallons per second.
100 Colorado miner's inches equal 130 California miner's inches.
100 Colorado miner's inches for one day equal 5.17 acre-feet.
100 United States gallons per minute equal 0.223 second feet.
100 United States gallons per minute for one day equal 0.442 acre-feet.
1,000,000 United States gallons per day equal 1.55 second-feet.
1,000,000 United States gallons equal 3.07 acre-feet.
1,000,000 cubic feet equal 22.95 acre-feet.
1 acre-foot equals 325,850 gallons.
1 inch deep on 1 square mile equals 2,323,200 cubic feet.
1 inch deep on 1 square mile equals 0.0737 second-foot per year.
1 inch equals 2.54 centimeters.
1 foot equals 0.3048 meter.
1 yard equals 0.9144 meter.
1 mile equals 1.60935 kilometers.
1 mile equals 1,760 yards; equals 5,280 feet; equals 63,360 inches.
1 square yard equals 0.836 square meter.
1 acre equals 0.4047 hectare.
1 acre equals 43,560 square feet; equals 4,840 square yards.
1 acre equals 209 feet square, nearly.
1 square mile equals 259 hectares.

1 square mile equals 2.59 square kilometers.
1 cubic foot equals 0.0283 cubic meter.
1 cubic foot equals 7.48 gallons; equals 0.804 bushel.
1 cubic foot of water weighs 62.5 pounds.
1 cubic yard equals 0.7646 cubic meter.
1 gallon equals 3.7854 liters.
1 gallon equals 8.36 pounds of water.
1 Imperial Gallon equals 1.20 U. S. Gallons.
1 gallon equals 231 cubic inches (liquid measure).
1 pound equals 0.4536 kilogram.
1 avoirdupois pound equals 7,000 grains.
1 troy pound equals 5,760 grams.
1 meter equals 39.37 inches. Log. 1.5951654.
1 meter equals 3.280833 feet. Log. 0.5159842.
1 meter equals 1.093611 yards. Log. 0.0388629.
1 kilometer equals 3,281 feet; equals five-eights mile, nearly.
1 square meter equals 10.764 square feet; equals 1.196 square yards.
1 hectare equals 2.471 acres.
1 cubic meter equals 35.314 cubic feet; equals 1.308 cubic yards.
1 liter equals 1.0567 quarts.
1 gram equals 15.43 grains.
1 kilogram equals 2.2046 pounds.
1 tonneau equals 2,204.6 pounds.
1 foot per second equals 1.097 kilometers per hour.
1 foot per second equals 0.68 mile per hour.
1 cubic meter per minute equals 0.5886 second-foot.
Acceleration of gravity equals 32.16 feet per second every second.
1 horse power equals 550 foot-pounds per second.
1 horse power equals 76 kilogram-meters per second.
1 horse power equals 746 watts.
1 horse power equals 1 second-foot falling 8.80 feet.
1⅓ horse power equals about 1 kilowatt.
To calculate water power quickly:

$$\frac{\text{Sec.-feet} \times \text{fall in feet}}{11} = \text{Net horse power on water wheel, realizing 80 per cent of the theoretical power.}$$

1 atm. equals 14.7 pounds square inch at sea level
1 atm. equals 33.947 feet water at 62° F.
1 atm. equals 30 inches Mercury at 62° F.
1 atm. equals 29.92 inches Mercury at 32° F.

1 atm. equals 760 mm. Mercury at 32° F.
1 atm. equals 1.033 kg. per sq. cm.
1 lb. per sq. inch equals 2.0416 ins. Mercury at 62° F.
1 lb. per sq. inch equals 2.0355 ins. Mercury at 32° F.
1 lb. per sq. inch equals 27.71 ins. Water at 62° F.
1 lb. per sq. inch equals 2.309 feet of water at 62° F.
1 lb. per sq. inch equals 0.0703 kg. per sq. cm.
1 foot of water at 62° F. equals .433 lbs. per sq. inch.
1 inch of Mercury at 62° F. equals .491 lbs. per sq. inch.
1 inch of Mercury at 62° F. equals 1.132 ft. of water at 62° F.

Duty of Water for Irrigation in the United States

The following table is taken from "Irrigation and Drainage" by Professor F. H. King of the University of Wisconsin:

Table showing the highest probable duty of water per acre for different yields of different crops:

Bushels Per Acre..	15	20	30	40	50	60	70	80	100	200	300	400
Name of Crop	Least Number of Acre-Inches of Water											
Wheat	4.5	6	9	12	15	18						
Barley	3.21	4.28	6.42	8.56	10.7	12.84	14.98	.....	.....	.....	.....	.....
Oats	2.35	3.13	5.70	6.27	7.84	9.40	10.98	12.54	15.68	.....	.....	.....
Maize	2.52	3.36	5.04	6.72	8.4	10.08	11.75	13.43	16.77	.....	.....	.....
Potatoes	.....	.41	.62	.83	1.03	1.24	1.45	1.65	2.07	4.14	6.2	8.27
Tons Per Acre....	1	2	3	4	6	8	10	12	14	16	18	20
	Least Number of Acre-Inches of Water											
Clover Hay 15 per cent water	4.43	8.85	13.28	17.7	26.55	35.4	44.25	.....	.....	.....	.....	.....
Corn with ears 15 per cent water	2.08	4.16	6.24	8.32	12.47	16.61	20.72	24.95	29.1	33.26	37.42	41.58
Corn silage 70 per cent water	1.41	2.82	4.23	5.64	8.46	11.28	14.1	16.92	19.74	22.56	25.38	28.2

The above table shows the minimum quantity of water required to produce a yield of 40 bushels of wheat per acre if dependent entirely upon irrigation to be 12 inches in depth of water per acre. This is equal to 43.560 cubic feet or 326.700 gallons per acre. This quantity would be distributed on the land at intervals in depths of from 3 to 4 inches at a time to suit the requirements of the growing crop.

King also gives the following

"It has been shown that under conditions in which no water can be lost by surface or under-drainage:

Clover uses 5.089 Acre-inches in producing one ton of dry matter
Oats uses 4.447 Acre-inches in producing one ton of dry matter
Barley uses 4.096 Acre-inches in producing one ton of dry matter
Maize uses 2.391 Acre-inches in producing one ton of dry matter
Potatoes use 3.399 Acre-inches in producing one ton of dry matter

"These figures are an approximate measure of the demands of those crops for water and if one, two or three tons of dry matter per acre are to be produced by these crops, then the amount of available rainfall needed will be given by multiplying the figures in this table by the yield which is expected per acre from the soil."

Open Ditches

A drainage ditch should be of sufficient capacity to flow only three-fourths full at flood height.

A channel with vertical sides offers least resistance to current, and if this form could be maintained, it would carry greatest volume of water in proportion to its cross section area. But since nothing but rocky material will stand in this form, most ditches are made trapezoidal in cross section. Ordinary clays will stand with a slope of 45° or 1 to 1. Loose loamy and sandy soils usually require slopes of 1½ to 1.

Ditches should have sufficient fall to make them self-cleaning. In soil and clay not easily displaced, this is about 4 feet per mile, which for ditches of ordinary size gives a mean velocity of 2⅜ miles per hour when running full. Increasing the depth of a ditch increases the head so that ditches of light grade channels should be made as deep as possible.

The greatest velocity of a stream is found in the thread of current in the center of the channel just below the surface. All other parts have a less velocity in proportion as they approach the bottom and sides of the channel. The mean velocity in a trapezoidal ditch is about four-fifths of the surface velocity and is that found at a point in the center line of the ditch a little more than half way from the surface to the bottom. The bottom velocity is about seven-tenths that of the surface.

The following table taken from "Engineering for Land Drainage", by Charles G. Elliott, shows the effect that increase in depth has upon the mean velocity in a rectangular channel 10 feet wide with a grade of 3 feet per mile.

Mean Velocity of Water at Different Depths in Rectangular Ditch 10 Feet Wide, Grade 3 Feet per Mile

Depth in Feet	Mean Velocity in Feet per Second
0.5	1.4
1.5	2.3
2.0	2.6
2.5	2.8
3.0	2.9
4.0	3.2
5.0	3.4
6.0	3.6
8.0	3.8

Relation of Breadth and Depth of Channel to Surface and Mean Velocity

The following table from "Fanning's Hydraulic Engineering, shows the relation of breadth, depth, surface velocity and mean velocity to each other for rectangular smooth channels when water is from 5 to 10 feet deep. Let b = breadth, d = depth, V = surface velocity and v = mean velocity.

When b = 2d then v = .920V
When b = 3d then v = .910V
When b = 4d then v = .896V
When b = 5d then v = .882V
When b = 6d then v = .864V
When b = 7d then v = .847V
When b = 8d then v = .826V
When b = 9d then v = .805V
When b = 10d then v = .780V

The mean velocity for a trapezoidal channel will be a little less and decreases as the sides slopes are flattened.

Safe Velocity of Flow in Ditches

The safe velocity is the highest velocity at which it is safe to allow water to flow in a ditch to prevent erosion or washing of banks. Ditches should be made as nearly self cleaning as possible and this often requires increasing the velocity of flow, particularly where water carries a large amount of silt, to practically the limit of safety.

Kents Eng. Hand Book gives the following:

Safe Velocity of Water in Ditches in Feet Per Minute

Soft Brown Earth	18 ft.
Soft Loam	36 ft.
Pure Sand	66 ft.
Gravel	156 ft.
Sandy Soil 15% clay	72 ft.
Sandy Soil 40% clay	108 ft.
Loamy Soil 65% clay	180 ft.
Clay Loam 85% clay	288 ft.
Agricultural Clay 95% clay	372 ft.
Clay	432 ft.

The quantity of water any ditch will convey safely depends on the gradient, the kind of soil, the depth of ditch, and whether cut to templet from solid earth or cut irregularly and banks allowed to cave and form a deflection of the stream from side to side.

Irrigation Quantity Tables

Amount of Water Required to Cover One Acre to Given Depths			Second Feet Reduced to Gallons and Acre Feet				Gallons required to cover a given number of acres to a depth of one foot. (Acre foot)	
Depth in Inches and Feet (Acre Inches and Acre Feet)	Cubic Feet (or Second Feet) Contained in one Acre to Depth Given in First Column	Gallons	Second Feet	Gallons per Minute	Gallons per Pumping Day of 12 Hours	Acre Feet per Pumping Day of 12 Hours	Acres (or Number of Acre Feet)	Gallons
1″	3630	27154	¼	112.2	80790	.2479	1	325851
2″	7260	54309	½	224.4	161579	.4959	2	651703
3″	10890	81463	¾	336.6	242369	.7438	3	977554
4″	14520	108617	1	448.8	323158	.9917	4	1303406
5″	18150	135771	1¼	561.0	403948	1.2397	5	1629257
6″	21780	162926	1½	673.2	484738	1.4876	6	1955109
7″	25410	190080	1¾	785.5	565527	1.7355	7	2280960
8″	29040	217234	2	897.7	646317	1.9835	8	2606812
9″	32670	244389	2½	1122.1	807896	2.4793	9	2932663
10″	36300	271542	3	1346.5	969475	2.9752	10	3258515
11″	39930	298697	4	1795.3	1292634	3.9669	15	4887772
1′ 00″	43560	325851	5	2244.2	1615792	4.9586	20	6512029
1′ 2″	50820	380160	6	2693.0	1938951	5.9503	25	8146285
1′ 4″	58080	434469	7	3141.8	2262109	6.9421	30	9775544
1′ 6″	65340	488777	8	3590.6	2585268	7.9338	40	13034058
1′ 8″	72600	543086	9	4039.5	2908426	8.9255	60	19551087
1′ 10″	79860	597394	10	4488.3	3231585	9.9173	80	26068116
2′ 00″	87120	651703	20	8976.6	6463170	19.8345	160	52136232

One cubic foot of water per second (exact 7.48052 gallons), constant flow is known as the "Second Foot". The "Acre Foot" is the quantity of water required to cover one acre to a depth of one foot.

Capacity of Standard Irrigation Ditches and Canals

The following table gives a slope of bank of one-half to one and is adapted only for irrigation ditches with low gradient and a sluggish flow. Drainage ditches require banks with slopes of from 1 to 1, to 1½ to 1, depending upon soil conditions and velocity.

Dimensions of Ditch				GRADE OF DITCH																					
				6 Inches Per Mile		1 Foot Per Mile		2 Feet Per Mile		3 Feet Per Mile		4 Feet Per Mile		5 Feet Per Mile		6 Feet Per Mile		7 Feet Per Mile		8 Feet Per Mile		9 Feet Per Mile		10 Feet Per Mile		Vol. Material to be removed per mile* cu. yds.
Top Width Feet	Bottom Width Feet	Depth Feet	Area of Cross Sect. Sq. Feet.	Velocity Feet Per Minute	Discharge Gallons Per Minute	Velocity Feet Per Minute	Discharge Gallons Per Minute.	Velocity Feet Per Minute	Discharge Gallons Per Minute	Velocity Feet Per Minute	Discharge Gallons Per Minute	Velocity Feet Per Minute	Discharge Gallons Per Minute	Velocity Feet Per Minute	Discharge Gallons Per Minute	Velocity Feet Per Minute	Discharge Gallons Per Minute	Velocity Feet Per Minute	Discharge Gallons Per Minute	Velocity Feet Per Minute	Discharge Gallons Per Minute	Velocity Feet Per Minute	Discharge Gallons Per Minute	Velocity Feet Per Minute	Discharge Gallons Per Minute	
1.5	1	0.5	0.625	10.8	45	20.4	94	30.0	139	36.6	171	42.6	197	48.0	229	54.6	256	57.6	269	62.4	292	65.4	305	70.2	467	122
3.0	2	1.0	2.5	21.0	386	28.8	543	43.2	812	52.2	1019	60.6	1135	69.6	1297	77.4	1450	82.2	1535	88.8	1660	93.0	1741	99.6	1863	489
4.5	3	1.5	5.625	27.6	1149	40.2	1692	60.0	2513	72.0	3016	83.4	3519	95.4	4026	106.8	4475	112.8	4739	121.8	5125	127.8	5381	136.8	5767	1100
6.0	4	2.0	10.	30.0	2244	51.0	3815	74.4	5565	89.4	6687	104.4	7809	119.4	8931	132.6	9918	140.4	10502	151.8	11351	159.6	11938	171.0	12790	1956
7.25	5	2.25	13.78	40.2	4142	57.6	5938	82.8	8536	101.4	10453	117.6	12118	130.8	13482	145.8	15026	157.8	16264	168.6	17378	178.8	18423	187.8	19357	2695
8.5	6	2.50	18.12	44.4	6014	63.6	8617	91.2	12360	112.2	15205	132.0	17889	147.0	19922	160.8	21794	174.0	23580	186.6	25205	196.8	26668	207.6	28135	3544
9.75	7	2.75	23.03	48.6	8370	69.0	11884	99.6	17158	122.4	21085	141.6	24388	157.8	27179	172.8	28364	187.2	32276	199.8	34414					4504
11.00	8	3.0	28.5	52.8	11256	75.6	16106	106.8	22768	130.8	27884	153.6	32740	171.6	36577	187.8	40033	202.8	45228							5573
12.25	9	3.25	34.5	55.8	14420	80.4	20766	115.8	29908	141.6	36573	163.8	42304	183.0	47263	200.4	51760									6752
13.50	10	3.50	41.12	60.0	18455	86.4	26573	122.4	37646	150.0	46128	171.6	52590	193.2	59421											8041

*In computing the volume of material in cubic yards per mile to be removed in the construction of each ditch, it is assumed that the ditches follow the surface of the ground, thus maintaining constant depth.

Losses from seepage and evaporation have not been taken into consideration. Experiments, however, show that losses from these sources are in direct proportion to surfaces exposed to the soil and to the air. It is best, therefore, to use as high a velocity of water in the ditch as the soil will permit.

NOTE—This table was revised from one given in Farmer's Bulletin No. 158, "How to Build Small Irrigation Ditches," published by U. S. Department of Agriculture, to which builder is referred for further instructions.

Velocity of Flow of Water in Feet per Second Through Various Sized Pipes

Diam. Pipe in Inches	U. S. GALLONS PER MINUTE													
	100	150	200	250	300	350	400	450	500	600	700	800	900	1000
3	4.539	6.808	9.078	11.347	13.616									
4	2.553	3.829	5.106	6.382	7.660	8.936	10.212	11.488	12.765					
5	1.634	2.451	3.268	4.085	4.902	5.719	6.536	7.353	8.170	9.804	11.438	13.072		
6	1.135	1.702	2.269	2.837	3.404	3.971	4.538	5.105	5.673	6.808	7 943	9.078	10.212	11.347
7	.834	1.250	1.667	2.084	2.501	2.917	3.335	3.752	4.168	5.002	5.836	6.669	7.503	8.336
8	.638	.957	1.277	1.596	1.914	2.233	2.553	2.872	3.191	3.829	4.467	5.105	5.744	6.382
9	.504	.756	1.009	1.261	1.513	1.765	2.017	2.269	2.521	3.026	3.530	4.035	4.539	5.043
10	.....	.613	.817	1.021	1.226	1.430	1.634	1.838	2.043	2.451	2.860	3.268	3.676	4.085
11	.....	.....	.675	.844	1.013	1.182	1.350	1.519	1.688	2.026	2.364	2.700	3.039	3.376
12	.....	.....	.567	.709	.851	.993	1.135	1.277	1.418	1.702	1.986	2.269	2.553	2.837
13	.....	.....	.....	.604	.725	.846	.967	1.088	1.209	1.450	1.692	1.934	2.176	2.417
14	.....	.....	.....	.521	.625	.729	.834	.938	1.042	1.251	1.459	1.667	1.876	2.085
15	.....	.....	.....	.....	.545	.635	.726	.817	.908	1.089	1.271	1.452	1.634	1.816
16	.....	.....	.....	.....	.....	.558	.638	.718	.798	957	1.117	1 276	1.436	1.596
18	.....	.....	.....	.....	.....	.....	.504	.567	.630	.756	.882	1.009	1.135	1.261
20	.....	.....	.....	.....	.....	.....	.....	.....	.511	.613	715	.817	.919	1.021
24	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.496	.567	.638	.709
30	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	. ....	.. ...	.454

Velocity of Flow of Water in Feet per Second Through Various Sized Pipes

Diam. Pipe in Inches	U. S. GALLONS PER MINUTE												
	1250	1500	1750	2000	2500	3000	4000	5000	7500	10000	15000	20000	25000
6	14.184												
7	10.421	12.505	14.589	16.673									
8	7.978	9.571	11.168	12.765									
9	6.304	7.566	8.827	10.087	12.609	15.130							
10	5.106	6.127	7.149	8.170	10.213	12.255	16.340						
11	4.220	5.064	5.908	6.752	8.440	10.128	13.504	16.880					
12	3.546	4.255	4.964	5.673	7.091	8.510	11.346	14.183					
13	3.022	3.625	4.230	4.834	6.043	7.252	9.669	12.086	18.129				
14	2.606	3.127	3.648	4.169	5.211	6.254	8.338	10.421	15.632				
15	2.268	2.724	3.177	3.631	4.539	5.447	7.262	9.078	13.616	18.155			
16	1.995	2.394	2.793	3.181	3.989	4.787	6.382	7 978	11 967	15.956			
18	1.576	1.891	2.206	2.521	3.152	3.782	5.043	6.303	9 455	12.607	18.912		
20	1.277	1.532	1.787	2.043	2.553	3.064	4.085	5.106	, 659	10.213	15.319		
24	.887	1.064	1.241	1.418	1.773	2.128	2.837	3.546	8.319	7.092	10.638	14.184	
30	.567	.681	.794	.908	1.135	1.362	1.816	2.269	3.404	4.539	6.809	9.078	11.347
36	.....	.....	.473	.630	.788	.946	1.261	1.576	2.364	3.152	4.728	6.304	7 880
42	.....	.....	.....	.463	.579	.695	.926	1.158	1.737	2.316	3.474	4.631	5.789
48	.....	.....	.....	.....	.443	.532	.709	.886	1.329	1.773	2.659	3.546	4.432

Flow of Water in Flumes

Velocity in feet per second, and quantity in gallons per minute
For various sizes and slopes.

	Slope in inches per 100 feet.	Cross Section in feet.	1 ft. x ½ ft.	1 ½ ft. x ¾ ft.	2 ft. x 1 ft.	3 ft. x 1 ½ ft.	4 ft. x 2 ft.	5 ft. x 2 ½ ft.	6 ft. x 3 ft.	7 ft. x 3 ½ ft.	8 ft. x 4 ft.	9 ft. x 4 ½ ft.	10 ft. x 5 ft.
Rough Lumber or Smooth Cement.	¾ in.	Velocity	1.2	1.6	1.9	2.6	3.2	3.7	4.2	4.6	5.0	5.4	5.8
		G. P. M.	270	825	1725	5200	11475	20850	33750	50250	72000	97500	130500
	1 ½ in.	Velocity	1.7	2.2	2.7	3.7	4.5	5.2	5.9	6.5	7.1	7.7	
		G. P. M.	382	1130	2430	7500	16200	29250	47625	71520	102000	140250	
	3 in.	Velocity	2.4	3.1	3.9	5.2	6.4	7.4					
		G. P. M.	540	1560	3510	10500	22950	41625					
	4 ½ in.	Velocity	2.3	3.8	4.8	6.4	7.8						
		G. P. M.	650	1920	4312	12900	28050						
	6 in.	Velocity	3.3	4.4	5.5	7.3							
		G. P. M.	742	2220	4950	14775							
Flumes of Smooth Planed or T. and G. Lumber.	¾ in.	Velocity	1.7	2.3	2.7	3.6	4.4	5.0	5.6	6.2	6.8	7.3	7.7
		G. P. M.	382	1162	2430	7275	15750	28125	45000	68250	97500	137500	172500
	1 ½ in.	Velocity	2.4	3.2	3.9	5.1	6.2	7.1	8.0				
		G. P. M.	540	1620	3510	10275	22350	39750	64500				
	3 in.	Velocity	3.4	4.5	5.5	7.2							
		G. P. M.	765	2250	4950	1455							
	4 ½ in.	Velocity	4.1	5.5	6.7								
		G.P.M.	915	2775	6037								
	6 in.	Velocity	4.8	6.4	7.7								
		G.P.M.	1072	3225	6937								

Condensers

SECTION THREE

Condensers

Condensers are used in connection with steam engines and steam turbines for the purpose of obtaining one of two important results, and generally both of them, viz.; a reduction in the consumption of steam, and the obtaining of more power from a given weight of steam.

To illustrate the action of a condenser, consider a simple idle steam engine. The atmosphere exerts a pressure of 14.7 pounds per square inch on both sides of the steam piston, and since the pressures are equal, they balance each other, and the piston cannot move. If the pressure of the atmosphere on one side of the piston is partially removed, the pressure on the opposite side will then be the greater, and the piston will move. Thus it will be seen that by removing the atmospheric pressure from one side of the piston, sufficient pressure may be obtained to move the piston without taking any steam from the boiler.

It is the office and duty of the condenser to remove a part of the atmospheric pressure from one side of the piston, thus enabling the higher atmospheric pressure on the other side to move the piston. If the engine will then run fast enough to develop, say ten per cent of its rated power, then the condenser will add ten per cent to the power of the engine, because thus far no steam has been taken from the boiler—we have simply removed a part of the atmospheric pressure or resistance to the piston.

When the atmospheric pressure is partially removed from one end of the cylinder, as soon as the piston reaches the end of the stroke, the higher pressure on the opposite side will act, and this will continue as long as the exhaust valve remains open, or through practically the whole stroke; it is the average pressure for the stroke, and consequently the difference between the high pressure on one side, and the low pressure on the other side, representing the mean effective pressure. This is the M. E. P. due to the condenser, and it will remain the same regardless of the steam pressure.

Now when we know the M. E. P. produced by the condenser, it is possible to find the horse power due to the condenser. Knowing the M. E. P. due to the condenser, and the M. E. P. produced by the steam, it is possible to find the horse power of the engine, when operating, non-condensing and condensing.

When the pressure in any engine is less than 14.7 lbs. per square inch (at sea level), a partial vacuum is said to exist. Thus when a condenser removes a part of the atmospheric pressure from one side of the piston, a partial vacuum will exist. A condenser (with its auxiliaries) produces a vacuum in two ways: first, by removing the air from the condensing chamber, exhaust pipe and cylinder; and second, by condensing the steam, which flows into it, thus causing it to return to a liquid—water.

If a cubic inch of water is evaporated at atmospheric pressure (14.7 lbs. absolute) or no pressure (by gauge), it will occupy 1642 cubic inches, and will displace 1642 cubic inches of air. If the steam thus produced be confined in an air tight receptacle, and condensed, it will return to water, and will then occupy $\frac{1}{1642}$ of the space formally occupied. Since no air can enter the receptacle, a vacuum will be produced. Thus if the exhaust pipe, cylinder, or condensing chamber be filled with steam, the air driven out, and if these parts are air tight and the steam be condensed, a vacuum will exist in the condenser, the exhaust pipe, and cylinder. The vacuum in a condensing engine is, therefore, produced by condensing the steam, and by removing the small amount of air that enters with the steam, together with what leaks in.

For non-condensing engines where the steam exhaust is direct to the atmosphere, at least the pressure of the atmosphere must act against the exhaust side of the piston, but in practice the back pressure is seldom less than 17 or 18 lbs. absolute, whereas, the back pressure acting against the exhaust side of the piston of a good condensing engine is often as low as 2 lbs. absolute.

Assume an engine is running without a condenser at a mean effective pressure of say 60 lbs. per square inch, and a condenser is added removing say 14 lbs. of back pressure; the mean effective pressure would be increased theoretically to 74 lbs., and the power of the engine would be correspondingly increased.

If the work done by the engine, after installing the condenser, remains the same, the mean effective pressure may be reduced again to 60 lbs. by cutting off the steam earlier in the stroke, and thus effecting a saving in steam consumption. Therefore, if the mean effective pressure produced by the condenser is known, one may readily estimate the horse power added by the condenser.

If it is desired to find the horse power added by the condenser, the following formula may be employed:

$$G = \frac{A \times R \times S}{33000} \qquad (35)$$

Where G = Gain in horse power due to the condenser,
A = Area of piston in square inches.
R = Reduction in back pressure in lbs. per square inch
S = Piston speed in feet per minute.

The reduction in back pressure R on the engine piston may be estimated as follows:

$$R = B - \left(\frac{30 - V}{2.04}\right) \qquad (36)$$

Where B = Absolute back pressure in pounds per square inch on the engine piston when running non-condensing, V = vacuum in inches of mercury (referred to a 30" barometer) produced by the condenser.

Generally speaking, a non-condensing engine requires from 20 to 30% more steam per horse power-hour, than a condensing engine of the same power. If a condenser be added to a non-condensing engine of say 250 H. P. running at 100 revolutions per minute, and the load and speed be kept the same after adding the condenser, the governor will produce an earlier cut-off, thus lowering the mean forward pressure of the steam, and the mean effective pressure will remain constant.

Suppose the cut-off occured at ½ stroke when running non-condensing, and at ⅜ stroke when running condensing, neglecting clearance, the saving in steam per stroke is the difference between ½ and ⅜, the piston displacement, or

$$\frac{.5 - .375}{.5} \times 100 = 25\%$$

Neglecting friction and other losses, the theoretical mean effective pressure may be determined as follows:

$$\text{M. E. P.} = P \times \frac{(1 + \text{HYP. log } r)}{r} - p \qquad (37)$$

Where
P = Absolute initial pressure in lbs. per square inch.
p = Absolute back pressure in lbs. per square inch.
r = Ratio of expansion =

$$\frac{\text{Length of stroke} + \text{Clearance}}{\text{Distance to cut off} + \text{Clearance}}$$

The following table shows the mean pressure per pound of initial pressure with different clearance and cut-offs.

Mean Pressure per Pound of Initial, with Different Clearances and Points of Cut-off

Cut-off in Fractions of the Stroke		PERCENTAGE OF CLEARANCE													
		0	1	1.5	2	2.5	3	3.5	4	4.5	5	5.5	6	6.5	7
1/10	0.1	0.3303	0.3439	0.3505	0.3568	0.3630	0.3690	0.3750	0.3808	0.3864	0.3919	0.3974	0.4027	0.4076	0.4126
1/8	0.125	0.3849	0.3966	0.4023	0.4078	0.4132	0.4187	0.4237	0.4287	0.4338	0.4386	0.4433	0.4480	0.4527	0.4571
1/6	0.167	0.4662	0.4757	0.4802	0.4844	0.4890	0.4933	0.4973	0.5014	0.5056	0.5096	0.5134	0.5173	0.5210	0.5245
1/6	0.188	0.5013	0.5097	0.5138	0.5181	0.5217	0.5259	0.5295	0.5332	0.5367	0.5405	0.5440	0.5474	0.5511	0.5546
1/5	0.20	0.5219	0.5298	0.5336	0.5376	0.5414	0.5449	0.5482	0.5517	0.5556	0.5588	0.5623	0.5656	0.5687	0.5716
1/4	0.25	0.5966	0.6025	0.6059	0.6090	0.6120	0.6148	0.6174	0.6207	0.6229	0.6258	0.6286	0.6312	0.6336	0.6359
3/10	0.30	0.6609	0.6663	0.6684	0.6712	0.6729	0.6755	0.6779	0.6803	0.6825	0.6845	0.6864	0.6882	0.6911	0.6927
1/3	0.333	0.6988	0.7029	0.7047	0.7076	0.7092	0.7106	0.7132	0.7144	0.7168	0.7190	0.7212	0.7219	0.7239	0.7257
3/8	0.375	0.7433	0.7458	0.7476	0.7494	0.7510	0.7525	0.7539	0.7569	0.7582	0.7593	0.7603	0.7630	0.7639	0.7646
2/5	0.40	0.7665	0.7691	0.7719	0.7729	0.7738	0.7765	0.7772	0.7778	0.7802	0.7806	0.7829	0.7831	0.7853	0.7874
1/2	0.50	0.8466	0.8484	0.8492	0.8503	0.8513	0.8522	0.8530	0.8539	0.8548	0.8556	0.8565	0.8573	0.8582	0.8590
3/5	0.60	0.9064	0.9076	0.9081	0.9087	0.9092	0.9097	0.9102	0.9107	0.9112	0.9117	0.9122	0.9127	0.9132	0.9136
5/8	0.625	0.9188	0.9194	0.9201	0.9206	0.9210	0.9215	0.9220	0.9224	0.9228	0.9233	0.9237	0.9241	0.9245	0.9249
2/3	0.667	0.9371	0.9378	0.9382	0.9385	0.9389	0.9392	0.9396	0.9399	0.9402	0.9405	0.9408	0.9411	0.9415	0.9418
7/10	0.70	0.9497	0.9502	0.9505	0.9508	0.9511	0.9513	0.9516	0.9518	0.9521	0.9524	0.9526	0.9528	0.9531	0.9533
3/4	0.75	0.9657	0.9661	0.9663	0.9665	0.9667	0.9668	0.9670	0.9672	0.9674	0.9675	0.9677	0.9679	0.9680	0.9682

Assuming a single non-condensing engine having a clearance of 5 per cent, and cutting off at ¼ stroke. Let the steam pressure at the throttle be 150 lbs. absolute and the back pressure 17 lbs. absolute. It is also assumed that the initial pressure in the cylinder is the same as the pressure at the throttle.

Referring to the column headed "5 per cent clearance," opposite ¼ cut off, the mean pressure per pound of initial pressure will be found to be equal to .6258. This multiplied by the initial pressure is

$$150 \times .6258 = 93.87 \text{ lbs.}$$

which is the mean forward pressure of the steam. Subtracting the absolute back pressure,

$$93.87 - 17 = 76.87 \text{ lbs. per square inch}$$

as the mean effective pressure on the piston.

Let it be required to find an approximate point of cut-off, which will maintain the same power of the engine, when running condensing with a vacuum of 26″, and it be understood that the speed and load, and consequently the initial and mean effective pressure remain the same.

Dividing 26″ of vacuum by 2.04 (1 pound pressure = 2.04 inches of mercury), gives 12.7 pounds per square inch, and subtracting this from the atmospheric pressure, leaves 2 pounds as the approximate absolute back pressure on the piston. The mean pressure ratio for the foregoing conditions may be found by adding the mean effective pressure to the absolute back pressure, and dividing by the absolute initial steam pressure.

Substituting the actual values:

$$\frac{76.87+2}{150} = .5258$$

as the mean pressure ratio required.

Referring again to the table and following down the column headed "5% Clearance", .5258 will be found to be between the values .5096 and .5405. Taking .5405 as the nearest figure in the table, it is found to correspond to a cut-off of $\frac{3}{16}$ or 18.8 per cent of the stroke.

The approximate saving in steam is:

$$\frac{.25-.188}{.25} \times 100 = 24.8 \text{ per cent.}$$

due to adding the condenser, and thereby shortening the cut-off.

If the saving in fuel is assumed to be in direct proportion

to the saving in steam, the condensing engine in this case will require 24.8 per cent less fuel, than the same engine would run ning non-condensing.

The increase in economy by the use of a condenser with a steam engine may be shown graphically as follows:

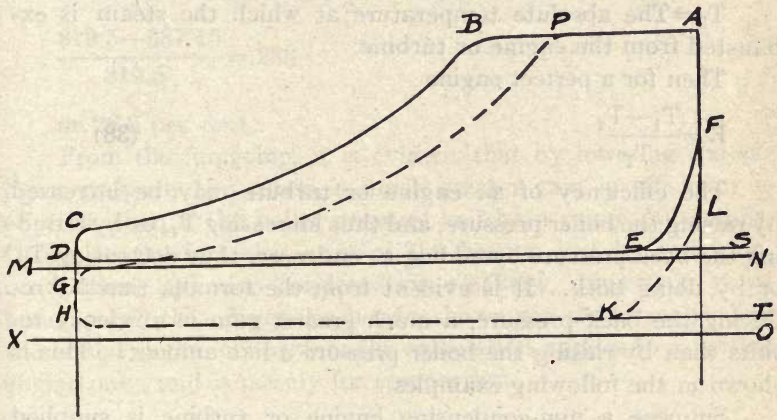


Fig. 74.

Let ABCDEF, figure 74, be the indicator card from a non-condensing engine. M-N is the atmospheric line, and O-X is the vacuum line. The back pressure as shown by the card, is O-S. The area of the card represents to some scale the work done per stroke. Now let a condenser be attached to the engine, and the back pressure will be lowered to O-T, the line H-K, instead of D-E now being the lower line of the card, and ABCHKL will be the new card, and its area as before, represents the work done per stroke. Hence by adding a condenser to the engine, the work done per stroke has been increased by an amount represented by the area FEDHKL, the steam consumption remaining the same.

Suppose the steam be cut off at a point P, making the area of the card APGHKL equal to the area of the original card ABCDEF; then the work per stroke is the same in both engines, but the condensing engine uses an amount of steam per stroke represented by the length A-P, while the non-condensing engine uses an amount of steam represented by A-B. Either case shows the economy of the condenser.

Gain in Thermal Efficiency

The gain in thermal efficiency due to adding a condenser to a steam engine or turbine, may be calculated as follows:

Let

E = Thermal efficiency of engine or turbine.

T_1 = Absolute temperature at which the steam is received by the engine.

T_2 = The absolute temperature at which the steam is exhausted from the engine or turbine.

Then for a perfect engine

$$E = \frac{T_1 - T_2}{T_1} \qquad (38)$$

The efficiency of an engine or turbine may be increased by raising the boiler pressure, and thus increasing T_1, or by reducing the back pressure by adding a condenser, thus decreasing T_2, or by doing both. It is evident from the formula that by reducing the back pressure, a much greater gain in efficiency results than by raising the boiler pressure a like amount. This is shown in the following examples.

Suppose a non-condensing engine or turbine is supplied with steam at 150 pounds absolute, as in the previous example, and exhausts at 17 pounds absolute, the absolute temperature T_1 corresponding to 150 pounds is (by referring to Steam Table in appendix):

358.5 + 461 = 819.5 degrees Fah.

The absolute temperature T_2, corresponding to 17 pounds absolute pressure, is:

219.4 + 461 = 680.4 degrees Fah.

Hence the thermal efficiency is:

$$E = \frac{819.5 - 680.4}{819.5} = .17$$

Or 17 per cent.

Now suppose the boiler pressure be raised from 150 pounds absolute to 200 pounds absolute, and the exhaust pressure kept the same, then the absolute temperature T_1 corresponding to 200 pounds absolute is 842.9 degrees Fahrenheit, and the absolute temperature T_2 corresponding to 17 pounds absolute is 680.4 degrees Fahrenheit. Hence the thermal efficiency is:

$$\frac{842.9 - 680.4}{842.9} = .193$$

or 19.3 per cent.

Suppose instead of raising the boiler pressure, the steam engine or turbine be connected up to a condenser, thus reducing the back pressure from 17 pounds to 2 pounds absolute. The absolute temperature T_2 corresponding to 2 pounds absolute is 587.15 degrees Fahrenheit, and the absolute temperature T_3, as already found, was 819.5 degrees Fahrenheit. In this case the thermal efficiency would be:

$$\frac{819.5—587.15}{819.5}=.283$$

or 28.3 per cent.

From the foregoing, it is evident that by lowering the exhaust pressure 15 pounds, a much greater gain in efficiency is effected, than if the boiler pressure had been raised 50 pounds. Owing to the fact that part of the heat is lost by radiation, conduction, cylinder condensation, leakage, imperfect valve action, etc., the efficiencies herein shown are never realized in the actual engine or turbine; the calculation applies to a perfect engine only, and is merely for comparison.

Most Economical Vacuum for Steam Engines

It has been definitely proven that the practical vacuum suitable for reciprocating engines is 26″, and there is no economy in carrying a higher vacuum. One reason for this is that in the reciprocating engine, it is practically impossible to take full advantage of a higher vacuum by expanding the steam in the cylinder down to the condenser pressure, on account of the excessively large volumes of steam produced by expansion to lower pressures. An attempt to do so would very largely increase the bulk of the engine and diminish its mechanical efficiency. By referring to the steam tables on pages 206–207, one can readily see how rapidly the volume increases for vacuums above 26″.

Even if it were found practical to expand to a lower pressure than 2 pounds absolute (26″ of vacuum), the low temperature of the exhaust would cool the cylinder walls to such an extent as to cause a very rapid increase in cylinder condensation.

The following table gives an idea of the steam consumption of steam engines operating condensing and non-condensing.

Pounds of Dry Steam per I. H. P. per Hour at Full Rated Load

I. H. P.	Simple High Speed		Simple Low Speed		Compound High Speed		Compound Low Speed	
	Non-Condensing	Condensing	Non-Condensing	Condensing	Non Condensing	Condensing	Non-Condensing	Condensing
10	65	50						
15	57	44						
20	52.5	40						
25	49	38						
30	46.5	36						
40	42.5	33						
50	40	30.2						
60	38	28.5						
75	35.5	26.2						
100	33	23.4	27	21.6	29.3	22.5	23.6	20
150	30.4	21.5	26.3	21	28.6	22	23.1	19.5
200	29.5	20.6	25.7	20.5	27.9	21.5	22.7	19
250	29	20.2	25.2	20	27.3	21	22.3	18.5
300	28.5	20	24.8	19.6	26.6	20.5	21.9	18.1
400			24.1	18.8	25.4	19.5	21.1	17.3
500			23.7	18.3	24.2	18.6	20.4	16.5
600			23.4	17.9	23.3	17.9	19.8	15.8
700			23.2	17.7	22.7	17.5	19.2	15.3
800			23	17.6	22.3	17.2	18.7	15
900			22.9	17.5	22.1	17	18.4	14.7
1000			22.8	17.4	22	16.9	18.2	14.5
1500								13.8
2000								13.5
2500								13.2
5000								12.5

ENGINEERING OF POWER PLANTS.

Most Economical Vacuum for Steam Turbines

For steam turbines, the average practice in this country as well as abroad is to employ a vacuum of 28″ referred to a 30″ barometer, although there are many installations under 1000 K. W., which operate on 26″ to 27″, and others of larger capacity, which operate on 28½″ to 29″ of vacuum.

There is a tendency on the part of steam turbine manufacturers, because of the obvious advantages of steam turbines over reciprocating engines for operating at high vacuums, to draw attention to the reduction in the steam consumption, when a plant is operated at high vacuum. The question is raised whether the actual economy takes into consideration the increas-

ed first cost of the condenser, pumps, and piping, together with the increased operating expense. The local conditions, particularly the temperature of the cooling water, determine this to a marked degree. In locations which are practically at sea level, and where the temperature of the water supply for the condenser is very low, it is doubtless profitable to install condensing apparatus of sufficient size to operate steam turbines at from 28½″ to 29″ of vacuum.

The following table gives the theoretical and practical vacuum at sea level for varying temperatures of cooling water.

Vacuum At Sea Level For Varying Temperatures of Cooling Water

Temperature of Cooling Water Deg. Fah.	Theoretical Possible Vacuum, Inches	Perfect Condenser No Temp. Difference Inches	Actual Condenser 16° Fah. Difference Inches	Actual Condenser 15° Fah. Difference Inches
Ratio Water to Steam	Infinite Ratio	60-1 Ratio	60-1 Ratio	100-1 Ratio
32	29.83	29.67	29.43	29.54
60	29.50	29.12	28.56	28.82
70	29.30	28.73	27.72	28.38
75	29.10	28.51	27.37	28.11

J. R. Bibbins

In modern surface condenser installations, there is usually a difference of about 15° Fah. between the temperature of the condensed steam and the discharge water. It will be seen then, in the above table, that with a reasonable ratio of cooling water to the steam of 60 to 1, the maximum vacuum obtainable when cooling water is taken at 60°, is 28.6″, and when taken in at 70°, is only 27.7″.

The following table shows the actual percentage of saving when the condensing equipment is increased so a plant can be operated at 28″ of vacuum, instead of 26″ of vacuum. It is estimated that the cost of the condensing equipment, including pumps and piping will be $4000.00 more for a 2000 K. W. plant to operate at 28″ of vacuum than at 26 inches of vacuum.

Relative Economy of 28" of Vacuum Over 26" of Vacuum for a 2000 K. W. Plant

Increased Cost of Equipment is $4000.00

Net Saving expressed as Percentage of Increased Capital Cost to Secure 28" of Vacuum over that for 26" of Vacuum	Average Load K. W.	Hours of Service per Day	Actual Evaporation, Pounds	Steam Consumption Average Lbs. per K. W. Hour	Water Saved per K. W. Hour by Raising Vacuum from 26" to 28"	Cost Coal, Dollars per Ton
118	1500	24	9.5	23	1.84	4.50
27	1000	24	8	22	1.76	2.25
4	1000	10	8	22	1.76	1.13

J. R. Bibbins

In the calculations, the rate of interest was taken at 5% and depreciation at 7½% on the extra equipment. The cost of extra power consumed was at the rate of 1 cent per K. W. hour and 10 cents per 1000 gallons of feed-water saved.

Although it may be stated in general it is profitable to equip a plant to operate under normal conditions at a vacuum of 28 inches, there are cases where there is practically no saving in so doing. In the third case given above where the plant is working on a ten-hour day, and coal is cheap, the gain is only 4 per cent.

Operation at 29" of vacuum compared with 28" is not so favorable to the higher vacuum as the comparison of 28" of vacuum with 26" of vacuum.

It will be observed in the following table that the volume of steam is increased practically in the same ratio (the volume is practically doubled) when the vacuum is increased from 28" to 29", as when increased from 26" to 28".

Table of the Volume of a Pound (Specific Volume) of Dry Saturated Steam at High Vacuums

Vacuum	Volume in cubic feet
29	665
28	342
26	176

In the above table, the ratio of the volume at 28" vacuum to that at 26" vacuum is 1.94, and the ratio of the volume at 29" of vacuum to that at 28" vacuum is 1.95.

The capacity of the condensing equipment for a steam turbine operating at 29″ vacuum must be practically four times as large as would be required for one exhausting at 26″ of vacuum. In other words, the volume of a pound of steam at the exhaust is 166 cubic feet larger at 28″ of vacuum than at 26″ of vacuum, but at 29″ of vacuum it is 323 cubic feet larger than at 28″ of vacuum. It has already been shown the cost of condensing equipment is $4000.00 more for a 2000 K. W. unit when 28″ of vacuum is substituted for 26″ of vacuum, and hence for 29″ of vacuum, the cost will be that much greater.

For turbine installations where the steam consumption is not reduced much more per inch of vacuum below 28-29″ than below 26-28″ in a comparison of economic operation at 29″ of vacuum with 28″, there is a high first cost for condensing equipment, which is not offset by a proportionate reduction in steam consumption, and there are very few places where an installation for operation at an average vacuum of 29″ is probable.

An idea of the relative quantity of the condensing water required for different vacuums may be gained by comparing that required for usual operating vacuums. For example, with injection water at 70° Fah., the usual temperature upon which condenser guarantees are based, it is customary to estimate that to obtain a vacuum of 26″ referred to 30″ barometer, approximately 26 pounds of water will be required for each pound of steam condensed, and approximately 2.5 times this quantity is required for a vacuum of 28″. With the injection water at 60° Fah., which should be considered the winter temperature, the quantity required for the foregoing vacuums is approximately 21 pounds and 39 pounds respectively. Knowing the quantity of the condensing water required, the cost of fuel, and the cost of water delivered to the condenser, the vacuum best suited to the condenser can be determined. Theoretically, the effect of operating a steam turbine by reducing the vacuum below that for which it is designed is to reduce the capacity and lower the rating at which the maximum economy is obtained.

The following table illustrates the percentage gain in economy per inch of vacuum for various vacuums. The close agreement between actual results and theoretical should be observed. This table applies to turbines using high steam pressures and superheat.

Inches of Vacuum	Gain in Per Cent			
	28″	27″	26″	25″
Curtis	5.1	4.8	4.6	4.2
Parsons	5.0	4.0	3.5	3.0
Westinghouse-Parsons	3.14	3.05	2.95	2.87
Theoretical	5.2	4.4	3.7	3.0

Mech. Eng. Feb. 24, 1906.

The economy in operating turbines condensing is well illustrated in the following tables, which give the results of tests on a 300 K. W. and a 500 K. W. turbine, when operating condensing, and non-condensing. The increase in steam consumption on the 300 K. W. turbine is about 50% and on the 500 K. W. turbine, about 75%, when running non-condensing.

Test of a 300 K. W. Turbine

K. W.	Steam Pressure Gauge	Superheat Degree Fah.	Vacuum Inches	Pounds K. W. Hour
303	158	0	26.58	23.15
297	161	0	0	34.20

Tests of a 500 K. W. Turbine

Brake Horse Power	Steam Pressure Gauge	Superheat Degrees Fah.	Vacuum Inches	Pounds B. H. P. Hour
383.5	152 2	.2	28.2	14.15
755.6	149.2	1.2	27.8	13.28
1121.9	148.8	5.1	26.5	14.32
385.6	148.2	2.7	8	24 94
766.8	147.3	2.6	.8	22.10
1144.4	126.1	11.4	.8	24.36

Steam Consumption of Small Steam Turbines

The majority of small steam turbines are generally operated non-condensing, and are rated on the brake horse power. Although the steam consumption for different size turbines may vary with the speed, a good idea of the steam consumption of non-condensing units may be secured from the following table:

Steam Consumption in Lbs. per B. H. P. per Hour at Full Load, 150 Lbs. Steam Pressure. Atmosphere Exhaust, No Back Pressure

Brake Horse Power	Pounds Steam B. H. P. per Hour
10	60
25	50
50	45
100	40
150	35
200	30
250	28

The following table gives the comparative figures for steam consumption for reciprocating steam engines and steam turbines.

Guaranteed Steam Consumption Lbs. per K. W. Hour, 150 Lbs. Steam Pressure

K. W.	Compound Corliss Steam Engine, Saturated Steam	100° Fah. Superheat	Steam Turbine 100° Superheat
	Vacuum 26″		Vacuum 28″
500	20.	18 5	18.1
1000	19.5	18.0	16.1
1500	18.5	17.0	15.8
2000	18.0	16.50	15.4

Water Rates of Turbo-Generators

60 CYCLE, 2300 VOLTS, A. C. GENERATORS

Condensing				Non-condensing	
Steam at 150 Lbs. Gauge, 28″ Vacuum		Steam at 200 Lbs. Gauge, 150° Superheat, 29″ Vacuum		Steam at 150 Lbs. Gauge Atmosphere Exhaust	
Rating K W.	Water Rate Lb. per Hour K. W., including Excitation	Rating K. W.	Water Rate Lb. per Hour per K. W. including Excitation	Rating K.W.	Water Rate Lb. per Hour per K.W.
50	32 to 42	4800	12 to 13.3	50	52 to 60
100	21.2 to 27.5	5000	11.9 to 13	100	43.5 to 50
200	19.5 to 25	7500	11.6 to 12.6	200	38.4 to 44
300	18.7 to 22.5	10000	11.2 to 12.3	300	36.5 to 42
400	18.2 to 21	12500	11.1 to 12	400	35.2 to 40
500	17.7 to 19.3	15000	11.1 to 11.8	500	34.1 to 38
600	17.4 to 19	17500	11 to 11.6	600	33.4 to 37
750	17 to 18.8	20000	11 to 11 4	750	32.7 to 36
1000	16.5 to 18.6	25000	10.9 to 11.3	1000	31.7 to 35
1250	16.2 to 18.3	30000	10.8 to 11.2	1250	31 to 34
1500	16 to 18	35000	10.7 to 11.1	1500	30.7 to 33
2000	15.7 to 17.7	40000	10.6 to 11.1	2000	30.1 to 32
2500	15.5 to 17.4	50000	10.5 to 11	2500	29.4 to 31
3000	15.4 to 17.1	60000	10.5 to 10.9	3000	28.9 to 30
3500	15.3 to 16.9	75000	10.4 to 10.8	3500	28.6 to 29

Loewenstein

Note.—Speeds 50-3500 K. W. inclusive, 3600 R. P. M., 4000 to 20000 K. W. inclusive, 1800 R. P. M., 25000 K. W. and over, 1200 R. P. M.

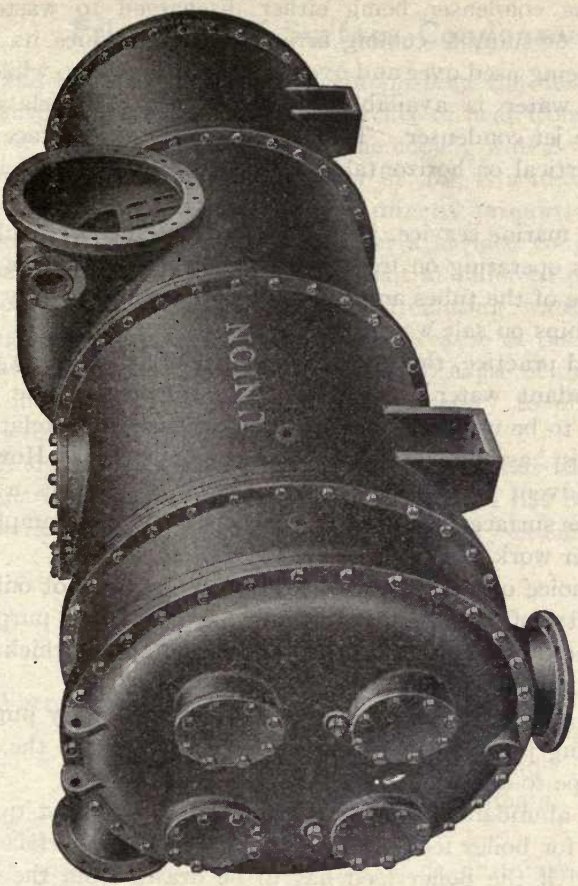


Fig. 75
Marine Type Surface Condenser.

Choice of Condensers

The choice of condensers depends almost entirely upon the local conditions, particularly the nature and quantity of boiler feed and cooling water available. For vacuums up to 26″, either the jet condenser or the surface condenser may be used. Where a high vacuum (26″ to 29″) is desired in connection with a steam turbine unit, a surface condenser is generally employed.

Broadly speaking, condensers may be divided into two classes: (a) those in which the cooling water is mixed with the steam, the combined quantities being discharged to the hot well; (b) those in which the cooling water and steam to be condensed are kept separate, the cooling water, after passing

through the condenser being either discharged to waste, or (by means of suitable cooling arrangement to reduce its temperature) being used over and over again continuously, while the condensed water is available for boiler feed. To class (a) belongs the jet condenser. To class (b) belong the surface condensers, vertical or horizontal, counter-current, or parallel-current.

In the marine service, jet condensers are employed only on board ships operating on fresh water. Surface condensers with water inside of the tubes and steam outside are universally used on board ships on salt water.

In land practice, the jet condenser is employed where there is an abundant water supply, which is sufficiently free from impurities, to be used for boiler feed on account of its relatively low first cost, as compared with surface condensers. However with the advent of the steam turbine, which requires a high vacuum, the surface condenser is being more and more employed in condenser work.

The choice of condenser will obviously depend not only on the quantity of the water available for condenser purposes, but also upon its quality, and the various conditions which arise in practice might be met as follows:

1. An abundant supply of fresh water, sufficiently pure for boiler-feeding purposes. The jet condenser would be the most suitable type to adopt.

2. An abundant supply of cheap water, but of a quality unsuitable for boiler feed. It is desirable to use a surface condenser, and if the boiler feed has to be drawn from the same source, some form of water softening or purifying plant; or, as the case of marine engines, an evaporator becomes necessary to make up the loss of feed water due to leakage.

3. A limited supply of water, of quality right for boiler feed, but of insufficient quantity to be discharged to waste, and relatively expensive to obtain. A jet condenser would be the best to adopt, and some form of water re-cooling plant becomes necessary, of sufficient capacity to reduce the temperature to a point low enough for continuous work.

4. A limited supply of water unsuitable for boiler feed. A surface condenser would be the most suitable type, combined with a water-cooling plant of ample capacity for continuous work, and with the water softening or purifying apparatus to make up the loss in boiler feed.

Principles of Surface Condensers

The important factor in accomplishing the desired results in a surface condenser is, a transference of heat from the steam through the walls of the tubes to the cooling water. The transference of heat per unit of area, or of size, is a measure of the efficiency of the apparatus, and is directly proportional to the temperature difference, or head. In a surface condenser, the temperatures of the fluids are different at different parts of the surface. The temperature of the circulating water increases during its passage through the tubes, because of the absorption of heat, and that of the steam decreases, because of the frictional drop in pressure. It is, therefore, necessary to obtain mean values for temperature differences.

A simple arithmetic mean is not correct, but the following formula developed mathematically by Grashof has been proven in practice to be very accurate, and is used very extensively.

$$D = \frac{T_2 - T_1}{\text{Hyp. Log.} \frac{T_s - T_1}{T_s - T_2}} \qquad (39)$$

Where D = Mean temperature difference
T_1 = The lowest temperature of the fluid
T_2 = The highest temperature of the fluid
T_s = The temperature of the gas or steam.

(See pages 206 and 207.)

In modern condenser practice, it is customary to make T_2 10° to 15° less than T_s, the temperature of the steam. This factor is dependent upon the design of the condenser.

Since the total heat to be abstracted in condensing one pound of exhaust steam is nearly constant within practical ranges of vacuum, it is apparent that the maintenance of high vacuums with temperatures rapidly approaching the temperature of the entering cooling water requires condensing equipment of much larger size proportionately than indicated by the vacuum.

Assuming the average temperature of the cooling water as 70° Fah.—heated to within 15° of the temperature of the entering steam, then a surface condenser capable of condensing 20,000 pounds of steam per hour at 26″ of vacuum will condense approximately only 14,000 pounds at a vacuum of 28″, although the number of heat units required per pound of steam

is practically the same in each case. The explanation of this lies in the value of the mean temperatures difference, which figures from the formula for the first case as 30.7° Fah. and for the last case as 22°.

The mean temperature difference requires for a maximum that the surfaces be arranged for counter-current flow, the water entering fartherest from the steam and passing consecutively through the tube nests, so as to finally pass out through the entering steam. This is the multi-pass condenser construction.

The transference of heat through a unit of condenser tube area per unit of mean temperature difference was early recognized as varying greatly under different conditions. The most apparent variation being an increase with an increase in the velocity of the cooling water. Many experimenters have carried out exhaustive tests along this line to determine the most practical value, but the results obtained vary greatly owing to the fact that in practice there are encountered certain resistances, which are in addition to the resistance offered by the metallic walls of the tubes.

The transference of heat produced by the temperature head is opposed by the resistance of the metallic walls of the tubes, the resistance of the steam side of the tube due to oil coating, or air-entrained steam, and the resistance on the water side of the tube due to the formation of scale.

Among the metals available for use as condenser tubes, copper is of the highest conductivity, and, furthermore, when properly alloyed, is less subject to corrosion than most others, thus permitting the using of thinner tubes. Hence all condenser tubes are a copper alloy. The size of tube is a determining factor in the thickness, larger tubes require greater thicknesses for mechanical strength, and from this view point small tubes are desirable.

To prevent the formation of a coating of oil on the tubes, which is detrimental to the heat flow, a high steam velocity must be maintained over the tubes, and there must be no dead ends or stagnant places in the condenser.

To eliminate the resistance due to air-entrained steam, surface condensers are generally arranged so that the steam sweeps the air ahead to the point of removal. By referring to figures 82-83, pages 202-203, the general arrangement of a modern high vacuum surface condenser can be seen, which

clearly shows the counter-current principle, as well as the location of the circulating-water. dry-air. condensate, and exhaust-inlet connections.

The resistance on the water side of the tube due to the formation of scale is very important, and too much attention cannot be paid to keeping the tubes clean. A high circulating water velocity will accomplish this to a marked degree, and is a more important reason for using small tubes, and several passes, than is generally recognized.

The coefficient of heat transmission or B. T. U. per square foot per degree difference per hour, is generally taken in practice at 300 to 400, depending upon the degree of vacuum, condenser design, etc.

Surface Condenser Calculations

Cooling Surface

A complete equation of the surface condenser is as follows:

$$S = \frac{W \times Q}{M \times U} \text{ or } U = \frac{W \times Q}{M \times S} \qquad (40)$$

U = B. T. U. per square foot per degree difference per hour.
M = Mean temperature difference degrees Fah.
W = Pounds of steam condensed per hour.
S = Square feet of cooling surface.
Q = Total heat removed by circulating water per pound of steam condensed (usually taken as 1000).

The table on page 204 gives the cooling surface required to condense 1000 lbs. of steam per hour under varying conditions. This table has been calculated by equation 40, and is based on a coefficient of heat transmission of 300 B. T. U. per square foot per degree difference per hour.

Cooling Water

In calculating the amount of cooling water required per pound of steam, the following practical equation may be used:

$$R = \frac{H - Q}{T_2 - T_1} \qquad (41)$$

H = Total heat of the steam (See pages 206-207)
Q = Heat of the liquid. (See pages 206-207)
T_2 = Final temperature of the cooling water.
(Generally 10°-15° less than T_s)
T_1 = The initial temperature of the cooling water.

The table on page 205 gives the cooling water required for surface condensers for vacuums of 25″ to 29″. This table has been calculated by equation 41 for cooling water temperatures of 50° to 85° Fah., and for temperature differences of 5° to 20° between the temperature due to the vacuum and the cooling water discharge temperature.

Size of Auxiliaries

In calculating the size of pumps to use with a surface condenser, the following gives a very good idea of customary practice.

Wet air pumps used in connection with surface condensers (without dry air pumps) for 26″ of vacuum and less, are given a displacement of 20 times the volume of steam condensed.

The condensate pump used in connection with a surface condenser (with a dry air pump), is generally given a displacement of 2 to 3 times the volume of steam condensed.

With regard to the capacity of the dry air pump, the following table will show what is considered good practice.

These figures are based upon an air tight system.

Vacuum	Air Pump Displacement per Pound of Steam Condensed
25	31
26	37½
27	48
28	55
28½	60
29	70

Example

Assume we have 500 K. W. steam turbine using 10,000 lbs. of steam per hour, and operating on 28″ of vacuum. It is required to find the size surface condenser, the amount of cooling water at 70° Fah., the displacement of the condensate pump, and the displacement of the dry vacuum pump for these conditions.

Solution

1000 lbs. steam per hour equals 2 gallons per minute.

10,000 lbs. of steam per hour equals 20 gallons per minute.

$T_1 = 70°$

$T_2 = 86.15°$ (Assume 15° lower than T_s.)

$T_s = 101.15$ (See page 206)

The mean temperature difference from equation 39, substituting the above values equals:

$$D = \frac{86.15 - 70}{\text{Hyp. log.} \frac{101.15 - 70}{101.15 - 86.15}}$$

$$= \frac{16.15}{\text{Hyp log. } 2.07}$$

$$= \frac{16.15}{.7275}$$

$$= 22.2$$

The surface of the condenser may now be calculated from equation 40 by substituting the values, and using for U a value of 300 B. T. U.

$$S = \frac{10{,}000 \times 1{,}000}{300 \times 22.2}$$

$$= 1500 \text{ square feet of cooling surface}$$

The amount of cooling water required at 70° Fah., is calculated from the equation 41 by substituting the values given

$$R = \frac{1104.1 - 69.12}{86.15 - 70}$$

$$= \frac{1034.98}{16.15}$$

$$= 64.1 \text{ pounds of cooling water per pound of steam.}$$

Now the amount of steam to be condensed by the example is 10,000 lbs. per hour, or 20 gallons per minute, so the amount of cooling water required to condense 10,000 lbs. of steam per hour will be

$20 \times 64.1 = 1282$ gallons per minute.

The displacement of the condensate pump from page 186, will be

$20 \times 3 = 60$ gallons per minute.

The displacement of the dry vacuum pump from page 186 (for 28″ of vacuum), will be

$$\frac{20 \times 55}{7.48} = 148 \text{ cubic feet.}$$

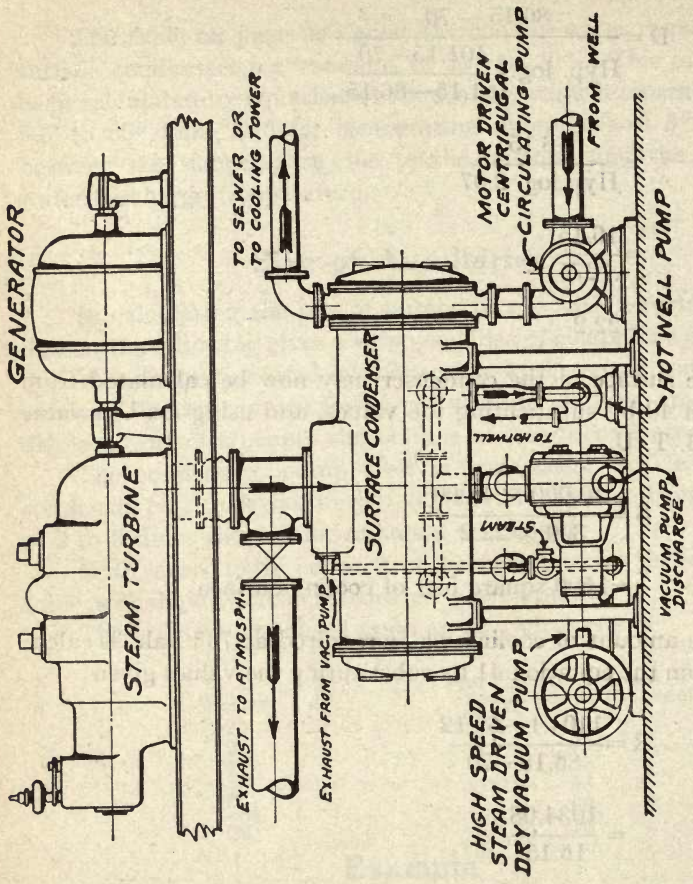


Fig. 76.
Steam Turbine Installation with Surface Condensing Equipment

Jet Condensers

The jet condenser consists of a combination of condensing chamber and pump. In this type of condenser, the condensing water and steam come in direct contact, and for this reason the jet condenser is particularly well adapted for use where the condensing water is suitable for feeding to the boilers. A sectional view of a Burnham jet condenser is shown in figure 77. The exhaust steam enters at A, and the condensing water at B. At D there is a cone-shaped spray nozzle connected with the tube C. The water issues from the nozzle D in an umbrella-shaped sheet or spray, which strikes the sides of the condensing chamber F. Thus the steam must pass through or into the spray on entering chamber F where it is condensed. The mixture of condensing water and condensed steam descends through

the contracted lower end of the condensing chamber F in a solid stream, which insures any remaining vapor being condensed, thence into the suction of the pump, which discharges the water through the valves T and opening J into the hot well. In addition to discharging the mixture of condensed steam and water, the pump removes any air that may enter in the injection water or through leaks. The pump also raises the injection water used for condensing the steam, the greatest lift being generally twenty feet. At E is a hand wheel with a long stem connected to the movable cone D, and by turning this wheel, the amount of injection water may be regulated to suit the requirements.

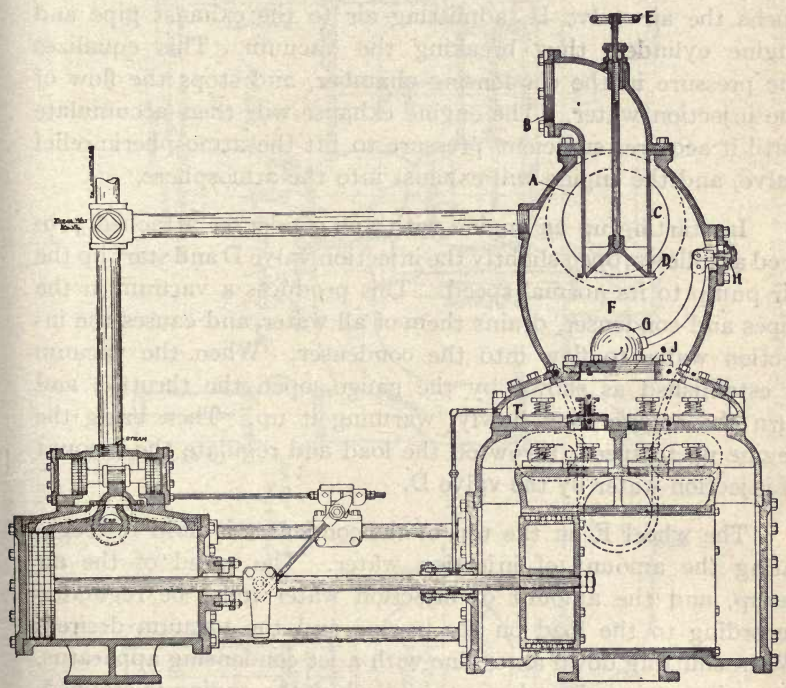


Fig. 77.
Sectional View Through Jet Condenser

The independent steam driven type of jet condenser as illustrated in figure 77 has the advantage of being absolutely independent of the main engine. It may be started before and stopped after the main engine, thus establishing a vacuum before the load is thrown on the engine, and draining the pipes and cylinder of the water of condensation and leakage. It may be run at any speed within reason, keeping the vacuum constant under changes of load.

To avoid the possibility of getting water over into the engine cylinder in case the pump stopped while the engine was running, a vacuum breaking device is arranged in the condensing chamber, as it is illustrated in figure 77. By referring to this figure, it will be observed that in case the pump slows down and stops, the water accumulating in the condensing chamber F will gradually lift the float G, and as the float rises, it in turn opens the air valve H, admitting air to the exhaust pipe and engine cylinder, thus breaking the vacuum. This equalizes the pressure in the condensing chamber, and stops the flow of the injection water. The engine exhaust will then accumulate until it acquires sufficient pressure to lift the atmospheric relief valve, and the engine will exhaust into the atmosphere.

In starting up an engine with a jet condenser attached, proceed as follows; open slightly the injection valve D and start up the air pump to its normal speed. This produces a vacuum in the pipes and condenser, drains them of all water, and causes the injection water to flow into the condenser. When the vacuum is established as shown by the gauge, open the throttle, and turn the engine over slowly, warming it up. Then bring the engine up to speed, throw on the load and regulate the amount of injection water by the valve D.

The wheel E on the top of the condenser is used for regulating the amount of injection water. The speed of the air pump, and the amount of injection water must be regulated according to the load on the engine and the vacuum desired. When shutting down an engine with a jet condensing apparatus, close the engine throttle first, and when the engine is stopped, and not until then, close the injection valve D, and lastly shut down the air pump. By shutting off the water supply before the air pump is stopped, the water already in the condenser and pipes is pumped entirely out, and there is no danger of it getting into the engine cylinder.

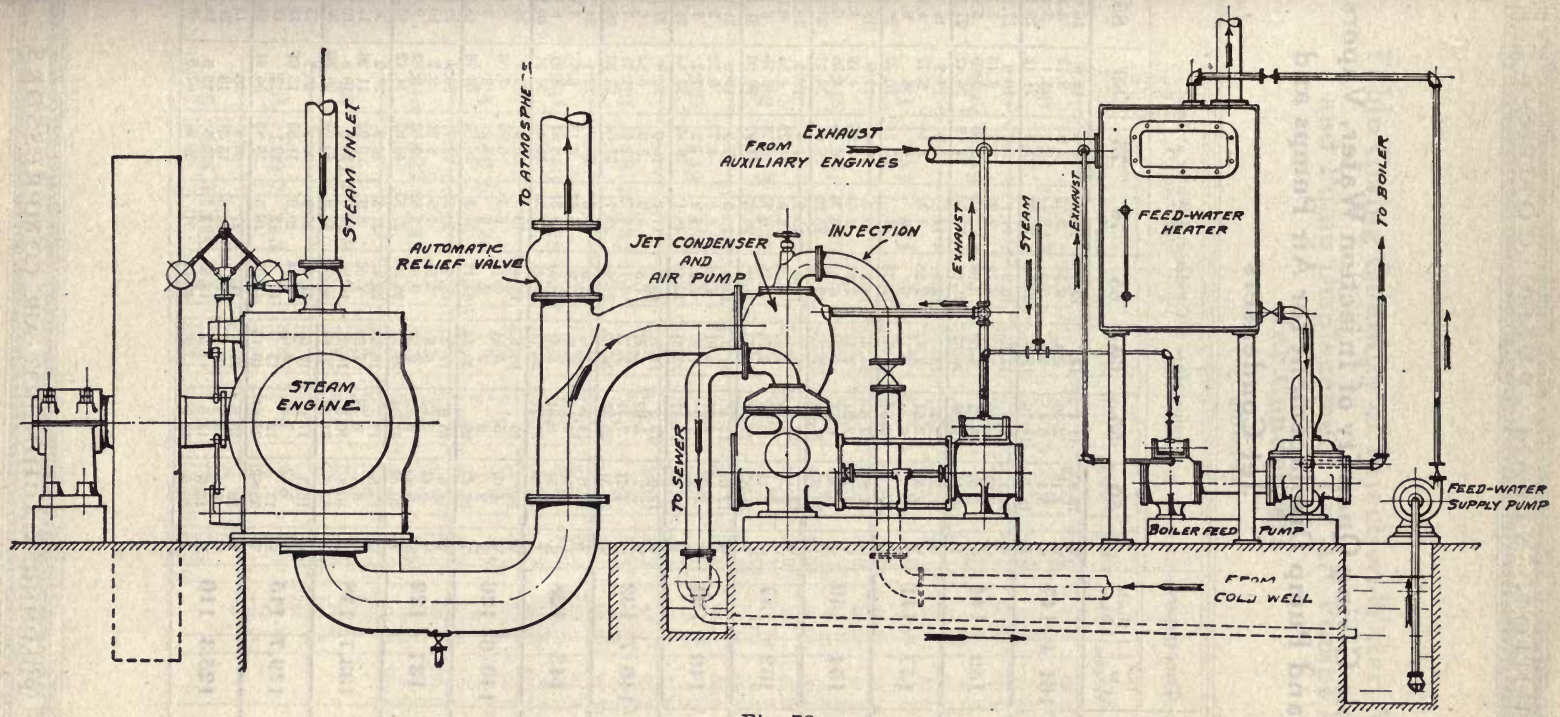


Fig. 78.
Steam Engine Installation with Jet Condenser Equipment.

Table Giving Quantity of Injection Water, Vapors and Pump Displacement for Air Pumps and Jet Condensers

Vacuum Based on 30" Barom.	Temperature		Temperature of Condensing Water at Inlet								
	Due to Vacuum	Disch. Water		50	55	60	65	70	75	80	85
20	161.4	147	W	10.8	11.1	12.	12.7	13.6	14.5	15.6	16.8
			V	2.16	2.26	2.4	2.54	2.72	2.9	3.12	3.3
			S	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4
			T	18.36	18.76	19.8	20.64	21.72	22.8	24.12	25 5
20½	159	144	W	11.1	11.8	12.5	13.2	14.1	15.2	16.3	17.7
			V	2.34	2.49	2.63	2.78	2.97	3.2	3.43	3.7
			S	5.69	5.69	5.69	5.69	5.69	5.69	5.69	5.6
			T	19.13	19.98	20.82	21.67	22.76	24.09	26.42	27.1
21	157	141	W	11.5	12.2	13	13.8	14.8	15.9	17.2	18.7
			V	2.5	2.64	2.8	2.98	3.2	3.44	3.72	4.0
			S	6	6	6	6	6	6	6	6
			T	20	20.84	21.8	22.78	24	25.34	26.32	28.7
21½	154	139	W	11.8	12.5	13.3	14.2	15.3	16.4	17.8	19.5
			V	2.78	2.94	3.13	3.34	3.64	3.86	4.19	4.5
			S	6.35	6.35	6.35	6.35	6.35	6.35	6.35	6.3
			T	20.93	21.79	22.78	23.89	25.29	26.61	28.34	30.4
22	152	137	W	12.1	12.6	13.7	14.6	15.7	17	18.3	20.2
			V	3	3.18	3.4	3.63	3.9	4.22	4.55	5.0
			S	6.75	6.75	6.75	6.75	6.75	6.75	6.75	6.7
			T	21.85	22.73	23.85	24.98	26.35	27.97	29.60	31.9
22½	149	134	W	12.6	13.4	14 3	15.3	16.5	17.9	19.6	21.6
			V	3.36	3.57	3.81	4.08	4.4	4.77	5.23	5.7
			S	7.2	7.2	7.2	7.2	7.2	7.2	7.2	7.2
			T	23.16	24.17	25.31	26.58	28.1	29.87	32.03	34.5
23	146.7	132	W	12.9	13.7	14.7	15.7	17.1	18.6	20.3	22.5
			V	3.7	3.92	4.21	4.5	4.9	5.33	5.82	6.4
			S	7.71	7.71	7.71	7.71	7.71	7.71	7.71	7.7
			T	24.31	25.33	26.62	27.91	29.71	31.64	33.93	36.6
23½	143	128	W	13.6	14.5	15.6	16.9	18.3	20	22.1	24.7
			V	4.18	4.46	4.8	5.2	5.63	6.15	6.8	7.6
			S	8.31	8.31	8.31	8.31	8.31	8.31	8.31	8.3
			T	26.09	27.27	28.71	30.41	32.24	34.46	36.21	40.6
24	140.6	126	W	14	15	16.1	17.4	19	20.9	23.1	26
			V	4.67	5	5.34	5.77	6.3	6.93	7.66	8.6
			S	9	9	9	9	9	9	9	9
			T	27.67	29	30.44	32.17	34.3	36.83	39.76	43.6
24½	137	122	W	14 8	15.9	17.2	18.7	20.5	22.7	25.4	28.9
			V	5.41	5.62	6.3	6.84	7.5	8.31	9.2	10.5
			S	9.82	9.82	9.82	9.82	9.82	9.82	9.82	9.8
			T	30.03	31.54	33.32	35.36	37.82	40.83	44.42	49.2
25	133.7	118	W	15.8	17	18.5	20.4	22.3	24.9	28.2	38.3
			V	6.3	6.78	7.38	8.06	8.9	9.93	11.26	15 2
			S	10.8	10.8	10.8	10.8	10 8	10.8	10.8	10 8
			T	32.9	34.58	36.68	39.06	42.0	45.63	50.26	56 2
25½	129.7	115	W	16.5	17.9	19.5	21.5	23.9	26.9	30.7	35 8
			V	7.32	7.94	8.65	9.54	10.6	11.93	13.61	15 8
			S	12	12	12	12	12	12	12	12
			T	35.82	37.84	40 15	43.04	46.5	50.83	56.31	63.6
26	125.3	110	W	18	19.6	21.6	24	27	30.9	36	43.2
			V	9	9.8	10.8	12	13.5	15.5	18	21.6
			S	13.5	13.5	13.5	13.5	13.5	13.5	13.5	13.5
			T	40.5	42.9	45.9	49.5	54	59.9	67.5	78.3

Table Giving Quantity of Injection Water, Vapor and Pump Displacement for Air Pumps and Jet Condensers

(*Continued*)

Vacuum Based on 30" Barom.	Temperature		Temperature of Condensing Water at Inlet							
	Due to Vacuum	Disch. Water		90	95	100	105	110	115	120
20	161.4	147	W	18.3	20.1	22.2	24.9	28.2	32.6	
			V	3.66	4.02	4.44	4.98	5.64	6.52	
			S	5.4	5.4	5.4	5.4	5.4	5.4	
			T	27.36	29.52	32.04	35.28	39.24	44.52	
20½	159	144	W	19.4	21.4	23.8	26.8	30.8	36.1	43.6
			V	4.08	4.51	5.01	5.64	6.49	7.6	9.18
			S	5.69	5.69	5.69	5.69	5.69	5.69	5.69
			T	29.17	31.60	34.50	38 13	43.98	49.39	58.47
21	157	141	W	50.6	22.8	25.6	29.1	33.8	40.3	50
			V	4.45	4.93	5.54	6.29	7.04	8.72	10.8
			S	6	6	6	6.	6	6	6
			T	31.05	33.73	37.14	41.39	46.84	55.02	66.8
21½	154	139	W	21.4	23.9	26.9	30.9	36.2	43.8	55.3
			V	5.04	5.62	6.33	7.27	8.52	10.31	13.01
			S	6.35	6.35	6.35	6.35	6.35	6.35	6.35
			T	32.79	35.87	39.58	44.52	51.07	60.46	74.66
22	152	137	W	22 4	25.1	28.5	32.9	39	47.9	61.9
			V	5.57	6.23	7 08	8.17	9.69	11.9	15.38
			S	6.75	6.75	6.75	6.75	6.75	6.75	6.75
			T	34.72	38.98	42.33	47.82	55.44	66.55	84.03
22½	149	134	W	24.	27.1	31	36.4	44	55.6	75.4
			V	6.4	7.23	8.27	9.71	11.73	14.83	20.11
			S	7.2	7.2	7.2	7.2	7.2	7.2	7.2
			T	37.6	41.53	49.47	53.31	62.33	77.63	102.71
23	146.7	132	W	25.2	28.6	33.1	39.2	48.1	62.2	88.2
			V	7.22	8.2	9.48	11.23	13.78	17.82	25.27
			S	7.71	7.71	7.71	7.71	7.71	7.71	7.71
			T	40.13	44.51	50.29	58.14	69.59	87.73	121.18
23½	143	128	W	27.09	32.2	37.9	46.2	59	81.7	132.7
			V	8.58	9.91	11.66	14.21	18.15	25.14	40.8
			S	8.31	8.31	8.31	8.31	8.31	8.31	8.31
			T	44.79	50.42	57.87	68.72	85.46	115.15	181 81
24	140.6	126	W	29.6	34.3	40.9	50.7	66.5	96.7	177.3
			V	9.82	11.37	13.56	16.81	22.05	32.06	58.8
			S	9.	9	9		9	9	9
			T	48.42	54.67	63.46	76.51	97.55	137.76	245.1
24½	137	122	W	33.4	39.6	48.5	62.8	89	152.6	
			V	12.22	14.49	17.74	22.98	32.56	55.83	
			S	9.82	9.82	9.82	9.82	9.82	9.82	
			T	55.44	63.91	76.06	95.60	131.38	218.25	
25	133.7	118	W	38.3	46.6	59.6	82.5	134		
			V	15.29	18.6	23.8	32.93	53.43		
			S	10.8	10.8	10.8	10.8	10.8		
			T	64.39	76.0	94.2	126.23	198.28		
25½	129 7	115	W	43.	53.8	71.7	107.5	215		
			V	19.03	23.86	31.8	47.68	95.35		
			S	12.	12	12	12	12		
			T	74.03	89.66	115.5	167.18	322.36		
26	125.3	110	W	54.	72	108	216			
			V	27.	36	54	108			
			S	13.5	13.5	13.5	13.5			
			T	94.5	121.5	175.5	337.5			

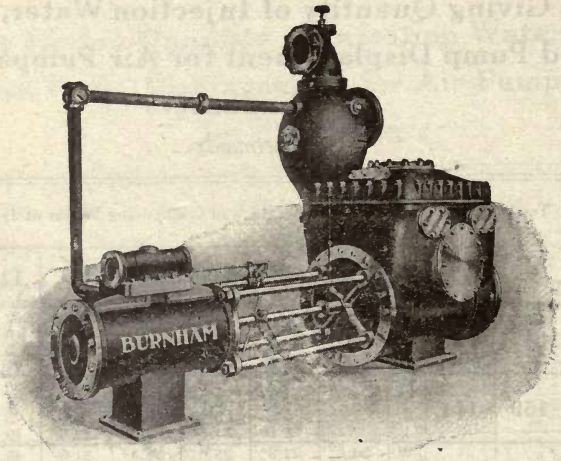


Fig. 79.
Burnham Air Pump and Jet Condenser

Jet Condenser Factors

The preceding table gives the quantity of injection water, the volumes of vapor with the steam and water, and the displacement required for the air pump with jet condensers for vacuums up to 26″, and using condensing water from 50 to 120° Fah. The figures are calculated from the formulae 42-43-44-45. All figures are stated in multiples of condensed steam.

In this table

W = Quantity of condensing water.
V = Volume of vapor from the water.
S = Volume of vapor from the steam and leaks.
T = (W + V + S) = Displacement of air pump.

Pumps for Jet Condensers

The size air pump to use with a jet condenser may be calculated by the following formulae:

$$D = W + V + S \qquad (42)$$

In which D = Displacement of pump.
W = Quantity of injection water.

$$V = \text{Volume of vapors from water} = \frac{2}{P_m} \times W \qquad (43)$$

$$S = \text{Volume of vapors from steam and leaks} = \frac{54}{P_m} \times Q \qquad (44)$$

P_m = Absolute pressure inches of mercury.
Q = Pounds of steam to be condensed per hour.

In calculating the amount of injection water, let H be the total heat in one pound of steam at the terminal pressure. This is assumed in practice at 1190 B. T. U.

t_s = Temperature of steam due to the vacuum.
t_1 = Temperature of injection water.
t_2 = Temperature of discharge water, which is assumed in practice 15° lower than the temperature due to the vacuum.

Each pound of injection water will be heated from t_1 to t_2, and the total heat absorbed by the water.

$$H_w = W\ (t_2 - t_1)$$

The heat given up by the steam condensing will be

$$H_s = Q\ (H - t_2)$$

Since the heat absorbed must equal that given up,

$$W\ (t_2 - t_1) = Q\ (H - t_2)$$

$$W = \frac{Q\ (H - t_2)}{t_2 - t_1} \qquad (45)$$

Example: Given 12000 pounds of steam per hour to be condensed, maintaining 26″ of vacuum, referred to 30″ barometer using 70° injection water. How much injection water is required? What size air pump is required?

Solution:

$Q = 12000$ lbs.

$t_1 = 70°$.

$t_s = 125°$ (see page 207)

$t_2 = 110°$

$H = 1190°$

Then substituting in equation 45,

$$W = \frac{12000\ (1190 - 110)}{110 - 70}$$

$$= \frac{12000 \times 1080}{40}$$

$= 324000$ pounds of injection water required per hour.

$$= \frac{324000}{8.3 \times 60} = 648 \text{ G. P. M.}$$

The volume of vapors from the water equals from formula 43,

$$V = \frac{2}{P_m} \times W = \frac{2}{4} \times 324000$$

$= 162000$ pounds per hour $= 324$ G. P. M.

The volume of vapors from the steam and leaks equals from formula 44,

$$S = \frac{54}{P_m} = \frac{54}{4} = 13.5 \times 12000$$

$= 162000$ pounds per hour $= 324$ G. P. M.

Hence from equation 42, the displacement of the air pump must equal

$D = 648 + 324 + 324 = 1296$ G. P. M.

Now referring to the table on page 197, you will find that the proper size air pump and jet condenser for these conditions is a 12x18x20, which has a condensing capacity of 12,300 pounds of steam per hour, and a displacement of 1322 G. P. M.

The following table gives the sizes of air pumps and jet condensers for various amounts of steam assuming 26″ of vacuum referred to 30″ barometer, and using cooling water at 50° to 80° Fah. The quantity of cooling water required for these conditions is also given.

Air Pumps and Jet Condensers

Size Pump	Strokes per Minute	Displacem't Gallons per Minute	Steam Condensed per Hour 26″ Vacuum, 30″ Barometer				Gallons per Minute Cooling Water Required			
			Temp. Cooling Water				Temp. Cooling Water			
			50°	60°	70°	80°	50°	60°	70°	80°
4½x 5x 8	100	68	840	740	630	500	31	32	34	36
4½x 6x 8	100	98	1200	1070	910	730	43	46	49	53
5 x 6x10	100	122	1500	1330	1130	910	54	58	61	66
5 x 7x10	100	166	2050	1810	1540	1230	73	78	83	89
6⅝x 8x10	100	217	2700	2380	2000	1610	98	103	108	116
6⅝x 9x10	100	275	3400	3000	2550	2040	122	130	138	148
6⅝x10x10	100	340	4200	3700	3160	2530	152	160	170	182
8 x10x12	100	468	5050	4450	3800	3000	180	192	205	217
8 x12x12	100	587	7250	6400	5450	4350	260	275	290	315
8 x12x16	75	587	7250	6400	5450	4350	260	275	290	315
10x14x16	75	800	10000	8750	7400	5950	360	375	400	430
10x16x16	75	1044	13000	11400	9750	7800	470	495	525	560
12x16x20	60	1044	13000	11400	9750	7800	470	495	525	560
12x18x20	60	1322	16400	14400	12300	9800	590	625	670	710
14x20x24	50	1632	20000	17700	15000	12000	720	765	810	870
14x22x24	50	1974	24400	21500	18300	14600	880	930	970	1060
14x24x24	50	2350	29200	25600	21700	17400	1050	1100	1170	1250
16x26x24	50	2758	34000	30000	25500	20300	1230	1300	1380	1470
16x28x24	50	3199	40000	35400	30000	23800	1440	1500	1620	1720
16x30x24	50	3672	45500	40000	34000	27300	1640	1720	1840	1970

Surface Condensers

In a surface condenser, figure 80, the steam to be condensed, and the cooling water do not come in direct contact with each other. The cooling water is circulated on the inside of a series of tubes, and the steam is condensed by coming in contact with the outside of the tubes. The condensed steam is drawn off by the air pump. The condensing water is drained, or forced through the tubes by the circulating pump. The external surface of the tubes which comes in contact with the steam is the condensing surface

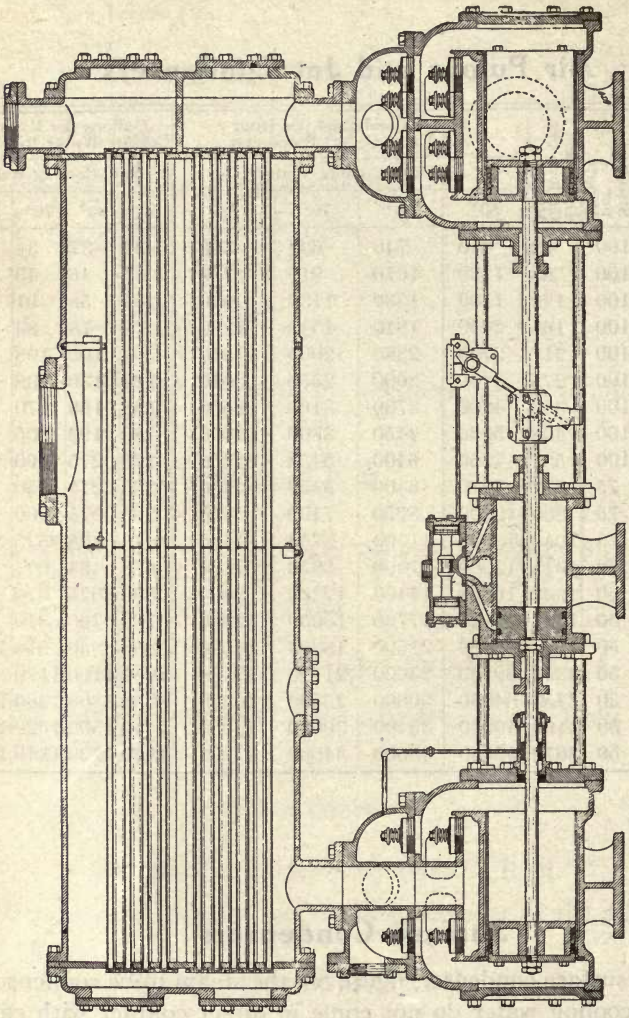


Fig. 80. Sectional View of Burham Air and Circulating Pump with Surface Condenser

Surface condensers usually consist of a cast iron shell, or casing, and it may be either cylindrical or rectangular in form. The cylindrical form is the simpliest and strongest for a given sectional area and weight of material, is cheaper to produce, and is considered more efficient than the rectangular type. The tube plates may be of cast iron, cast brass, or Muntz metal, depending upon the installation. In the marine service, condensers are always fitted with Muntz metal tube heads.

The tubes are seamless drawn brass (generally 60 copper and 40 zinc), and are usually made ¾" external diameter, and No. 18-B. W. G. in thickness, although ⅝" external diameter tubes, and either 18 B. W. G. or 20 B. W. G. are sometimes used.

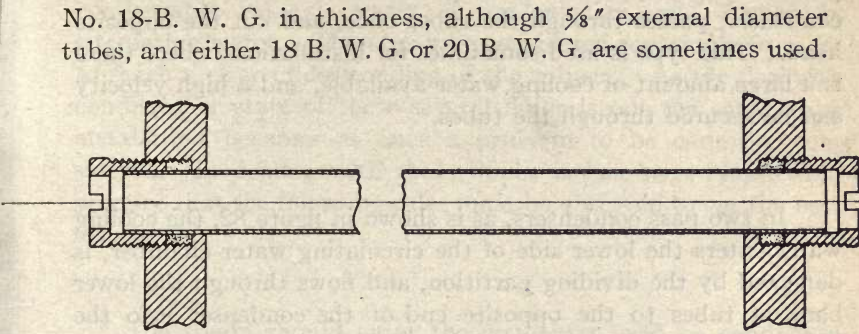


Fig. 81.

The tubes are secured in the tube plates usually by means of screwed ferrules and tape packing. Figure 81 illustrates the customary method of securing the tubes in the tube plates by screwed ferrules. It will be noticed the ferrules are provided with internal lips to prevent the displacement or creeping of the tubes by giving them ample room to expand or contract. Sometimes the tubes are expanded in the tube plates, but this method is not recommended for the reason that a certain amount of expansion and contraction will take place, which tends to produce slackness, and when a tube has become slack, it is a difficult matter to make it tight again.

The condenser shell is provided with one, and sometimes two circulating water chambers, depending upon the number of passes in the condenser. Suitably arranged cast-in partitions in the circulating water chamber and heads provide for the efficient circulation of the cooling water through the condenser tubes.

Surface condensers are built in the horizontal or vertical types, and may be arranged for the passage of the cooling water through the tubes with the exhaust steam surrounding them, or, as it is often done in water-works practice, with the steam passing through the tubes, and the cooling water on the outside.

A baffle, which is provided opposite the main exhaust steam inlet opening, prevents the steam from eroding the outer row of tubes, and deflects it in its path through the condenser.

In high vacuum surface condensers, drain plates are frequently provided to intercept the condensed steam flowing through the condenser, and deflect it to the sides and bottom of the condensing chamber, so as to keep the tubes dry.

Surface condensers are sometimes made with a single pass. In this type, the cooling water enters one end of the condenser, passes through the tubes once, and out the opposite head. This type is used sometimes for installations where there is a large amount of cooling water available, and a high velocity can be secured through the tubes.

In two pass condensers, as is shown in figure 82, the cooling water enters the lower side of the circulating water chamber, is deflected by the dividing partition, and flows through the lower bank of tubes to the opposite end of the condenser into the dished head, and thence returns through the upper bank of tubes and on the top of the circulating water chamber.

Surface condensers are made with as many as three and sometimes more passes, depending upon conditions.

Modern surface condensers are generally designed and installed so that the steam to be condensed enters horizontally at the side near the top, or vertically at the top of the shell, and in condensing flows downward, while the cooling water is introduced at the bottom, and leaves at the top, thus creating a counter-flow of the two fluids.

The condensate is removed from the bottom of the condenser, and with high-vacuum condensers using a dry-air pump, the non-condensable vapors are generally drawn off from the side of the condenser.

The primary functions of a surface condenser are to reduce the back pressure on the exhaust side of a steam prime mover; to conserve and return to the boiler the water of condensation, which is chemically pure feed water; to conserve and return to the boiler as many heat units as possible; and to remove from the feed water, the air in solution, thus avoiding pitting of the boilers.

To accomplish these results, the surface condenser must handle four separate fluids: steam, air (including other non-condensable vapors), water of condensation, and cooling or circulating water. These may be considered separately in order to reach a clear understanding of the subject. As the desirable condition or state of these several fluids is not the same, each installation becomes at once a problem to be carefully considered. In dealing with these fluids, it has been found from practice that the following rules must be observed to get the best results.

The steam should enter the condenser, and be conducted freely to all parts of the same with the least possible resistance; it should be reduced to the lowest practicable temperature, and should be converted into water for easy removal.

Air which is a nonconductor of heat should be rapidly cleared from the heat transmitting surfaces, collected at suitable places after being freed from the entrained water and vapor, and cooled to a low temperature of removal at a minimum volume, and consequently a minimum expenditure of energy.

The condensate should also be rapidly cleared from the heat transmitting surfaces, freed from the air, collected at suitable points for removal, and returned to the boiler at the maximum temperature.

The circulating water should pass through the condenser with the least friction, deposit a minimum amount of precipitate chemicals and absorb a maximum amount of heat.

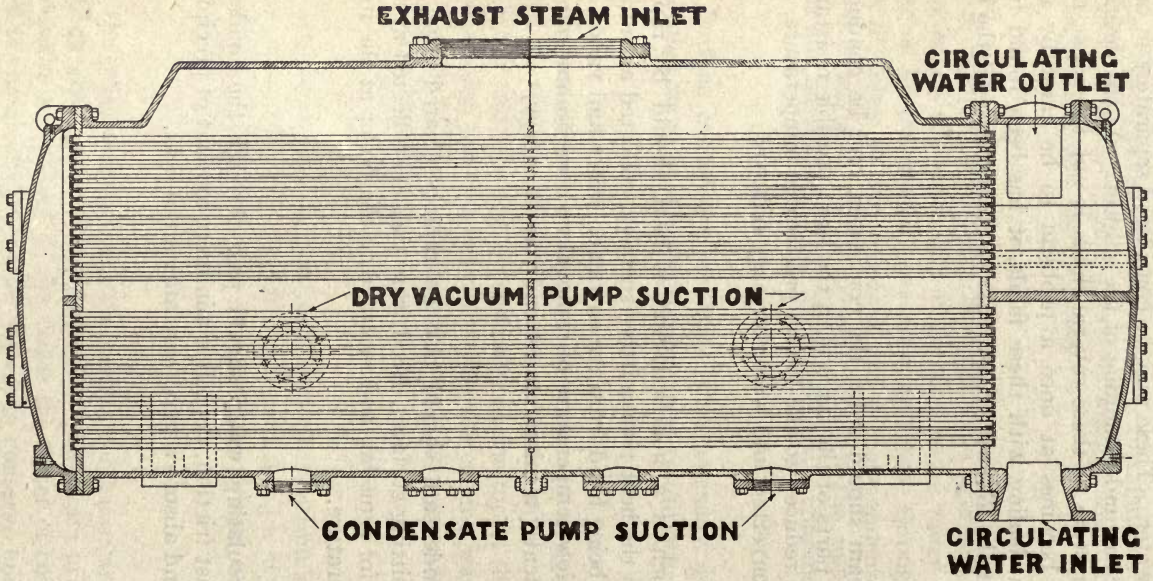


Fig. 82

Sectional View of a Two Pass High Vacuum Surface Condenser.

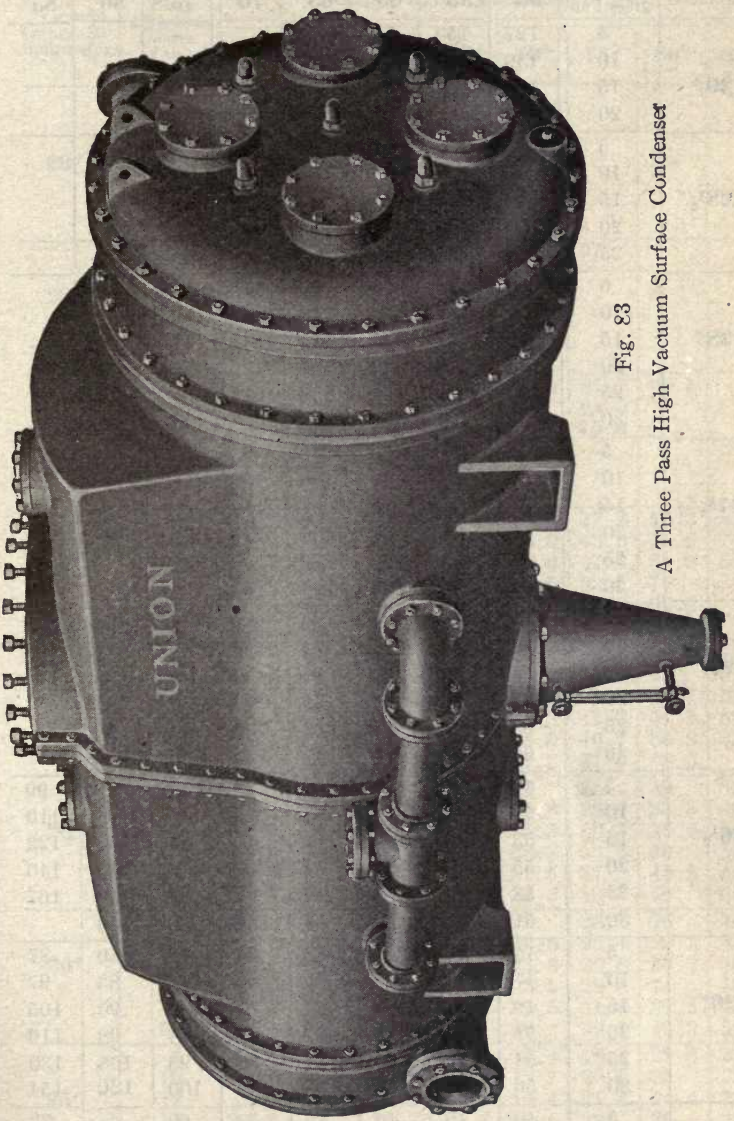


Fig. 83
A Three Pass High Vacuum Surface Condenser

Surface to Condense 1000 Pounds Steam Per Hour

Vacuum Inches, 30″ Barometer	Temp. Rise in Water Degrees Fah.	Inlet Water Temperature Degrees Fah.							
		50°	55°	60°	65°	70°	75°	80°	85°
29″	5	127	153	205	295				
	10	139	181						
	15	160	220						
	20	196							
28½″	5	89	100	116	137	170			
	10	90	105	125	153	202			
	15	98	115	140	180				
	20	108	129	164	228				
	25	120	150	203					
28″	5	70	75	83	99	116	144	186	
	10	72	82	93	108	129	162		
	15	78	87	101	119	149	191		
	20	84	95	111	135	172			
	25	90	104	125	159	218			
	30	99	116	145	198				
27½″	5	57	64	70	81	93	105	126	158
	10	61	69	77	85	100	116	143	
	15	65	73	81	93	110	130	163	
	20	69	77	88	102	121	150	197	
	25	74	83	96	112	138	180		
	30	79	90	106	128	165			
27″	5	51	58	63	70	80	87	104	121
	10	55	61	66	74	84	96	112	135
	15	58	64	71	79	90	104	124	154
	20	62	68	75	85	97	115	141	181
	25	65	72	81	92	108	130	166	
	30	68	77	88	101	122	155		
26½″	5	49	53	57	63	69	80	87	99
	10	51	55	61	66	73	83	94	110
	15	53	58	63	71	78	89	102	122
	20	55	61	67	74	84	98	115	140
	25	58	64	74	80	93	108	130	167
	30	61	69	79	87	100	120	153	
26″	5	45	50	54	58	63	70	80	87
	10	47	51	55	61	66	74	83	97
	15	49	55	59	63	71	80	90	105
	20	51	54	61	67	74	84	98	116
	25	54	59	65	71	80	93	108	130
	30	56	62	69	77	88	100	120	154
25″	5	40	45	46	51	53	60	64	73
	10	43	46	50	53	58	62	68	77
	15	44	47	51	55	60	65	72	82
	20	45	49	53	57	62	69	78	87
	25	47	51	55	60	66	74	83	96
	30	50	54	58	63	71	80	90	105

Pounds of Cooling Water per Pound of Steam Condensed.

Vacuum Inches, 30 Barometer.	Temp. difference Degree Fah.	Cooling Water Temperature Deg. Fah.							
		50°	55°	60°	65°	70°	75°	80°	85°
29″	6	45.7	58.3	80.8					
	10	55.1	75						
	12	61.8	87.5						
	15	75							
	18	95							
	20								
28½″	6	29.2	33.9	40.3	50	66.2	97		
	10	32.8	38.9	47.8	62.2	89.0			
	12	35	42	52.5	70.6				
	15	38.9	47.8	62.2	89.0				
	18	43.8	55.3	76					
	20	47.8	62.2	88.9					
28″	6	23	25.8	29.5	34.5	41.3	51.7	69.5	
	10	25.2	28.8	33.3	39.5	49	64.5		
	12	26.5	30.2	35.5	43	54.5	73.5		
	15	28.8	33.3	39.6	49	64.1			
	18	31.2	36.6	44.7	57	79			
	20	33.3	40	49	64.5				
27½″	6	19.6	21.6	24.5	27.6	31.8	37.2	45.3	58.3
	10	21.2	23.5	26.6	30.5	36.0	43.5	55.0	75
	12	22.2	24.7	28.2	32.5	38.6	47.5	61.8	88.0
	15	23.9	26.7	30.6	36.0	43.5	55.1	75	
	18	25.3	28.8	33.5	40.3	50	65.7	96.0	
	20	26.7	30.6	36.0	43.5	55.0	75		
27″	6	17.8	19.2	21.0	23.2	26.4	30.3	35.5	42.9
	10	18.7	20.6	22.8	25.8	29.4	34.3	42	51.3
	12	19.4	21.4	24.0	27.1	31.2	36.7	44.7	57.0
	15	20.6	22.8	25.7	29.3	34.3	41.2	51.5	68.5
	18	21.8	24.5	27.8	32.2	38.0	46.7	60.8	85.5
	20	23.0	25.7	29.4	34.3	41.2	51.5	68.5	
26½″	6	16	17.3	19	21	23	26	30	35
	10	17	18.5	20.7	23	25.8	29.6	33.5	40
	12	17.5	19.2	21.3	23.5	26.5	30.5	35.6	43.5
	15	18.5	20.3	22.5	25.2	28.8	33.5	40	50
	18	19.6	21.5	24	27.3	31.5	37	45.3	58.3
	20	20.3	22.5	25.2	28.7	33.5	40	50	66
26″	6	14.8	16	17.2	19	21	23	26	30
	10	15.9	17.0	18.8	20.6	22.6	25.4	29.2	33.7
	12	16	17.5	19.2	21	23.8	26.8	30.8	36.7
	15	17.0	18.5	20.4	22.6	25.4	29.0	34	40.3
	18	17.8	19.5	21.6	24.4	27.5	31.8	37.5	45.0
	20	18.5	20.3	22.7	25.3	29.	33.7	40.3	50
25″	6	13.4	13.9	14.9	16.2	17.4	19.3	21.3	23.5
	10	13.6	14.6	15.7	17.1	18.7	20.6	23	26
	12	14.1	15	16.3	17.6	19.4	21.5	24.4	27.6
	15	14.6	15.7	17.3	18.7	20.6	23	26	30
	18	15.3	16.5	18	19.8	22	24.7	28.5	33.0
	20	15.7	17.1	18.7	20.6	23	26	30	35

*Temperature difference between that due to saturated steam at any particular vacuum, and the temperature of the cooling water leaving the condenser.

Properties of Saturated Steam at Pressures Less than that of the Atmosphere

Vacuum Inches Hg. Referred to 30″ Barometer (Mercury at 58.4° F.)	Absolute Pressure in Inches Mercury at 32° F.	Temp. Degrees Fahrenheit	Specific Volume Cu. Ft. per Pound	Heat of the Liquid	Total Heat of Steam	Absolute Pressure in Lbs. per Sq. Inch
29.8	0.199	34.42	3004.0	2.43	1074.4	0.097
29.7	0.299	44.91	2040.0	12.97	1079.2	0.146
29.6	0.398	52.60	1554.0	20.68	1082.5	0.195
29.5	0.498	58.77	1259.0	26.85	1085.3	0.244
29.4	0.598	63.86	1063.0	31.93	1087.5	0.293
29.3	0.698	68.33	918.0	36.40	1089.6	0.342
29.2	0.797	72.27	810.0	40.32	1091.3	0.390
29.1	0.897	75.84	724.0	43.88	1092.9	0.439
29.0	0.997	79.07	657.0	47.11	1094.3	0.488
28.9	1.097	81.97	599.3	50.00	1095.6	0.537
28.8	1.996	84.61	552.5	52.63	1096.8	0.586
28.7	1.296	87.10	512.2	55.11	1097.9	0.635
28.6	1.396	89.47	476.9	57.47	1098.9	0.684
28.5	1.495	91.70	446.2	59.70	1100.0	0.732
28.4	1.595	93.97	419.6	61.78	1100.9	0.781
28.3	1.695	95.78	396.0	63.77	1101.7	0.830
28.2	1.795	97.67	375.0	65.65	1102.6	0.879
28.1	1.894	99.45	356.4	67.42	1103.4	0.928
28.0	1.994	101.15	339.6	69.12	1104.1	0.977
27.9	2.094	102.79	324.1	70.75	1104.8	1.026
27.8	2.194	104.35	310.3	72.31	1105.5	1.075
27.7	2.293	105.85	297.6	73.80	1106.1	1.123
27.6	2.393	107.30	286.0	75.25	1106.8	1.172
27.5	2.493	108.70	275.2	76.64	1107.4	1.221
27.4	2.592	110.05	265.1	77.99	1108.0	1.270
27.3	2.692	111.36	255.8	79.30	1108.6	1.319
27.2	2.792	112.63	247.2	80.56	1109.1	1.368
27.1	2.892	113.87	239.2	81.80	1109.6	1.417
27.0	2.991	115.06	231.9	82.98	1110.2	1.465
26.9	3.091	116.20	224.6	84.12	1110 7	1.514
26.8	3.191	117.32	218.0	85.14	1111.2	1.563
26.7	3.291	118.42	211.7	86.33	1111.7	1.612
26.6	3.390	119.50	205.8	87.41	1112.2	1.661
26.5	3.490	120.55	200.2	88.46	1112.6	1.710
26.4	3.590	121.55	195.1	89.46	1113.0	1.759
26.3	3.690	122.54	190.0	90.44	1113.4	1.808

Properties of Saturated Steam at Pressures Less than that of the Atmosphere—*Continued*

Vacuum Inches Hg. Referred to 30" Barometer (Mercury at 58.4° F.)	Absolute Pressure in Inches Mercury at 32° F.	Temp. Degrees Fahrenheit	Specific Volume Cu. Ft. per Pound	Heat of the Liquid	Total Heat of Steam	Absolute Pressure in Lbs. per Sq. Inch
26.2	3.789	123.51	185.5	91.41	1113.9	1.856
26.1	3.889	124.45	181.0	92.35	1114.3	1.905
26.0	3.989	125.38	176.7	93.28	1114.7	1.954
25.9	4.088	126.28	172.7	94.18	1114.9	2.003
25.8	4.188	127.17	168.9	95.06	1115.3	2.052
25.7	4.288	128.04	165.1	95.93	1115.7	2.101
25.6	4.388	128.90	161.5	96.79	1116.1	2.150
25.5	4.487	129.75	158.1	97.64	1116.5	2.198
25.4	4.58	130.59	154.8	98.48	1116.9	2.24
25.3	4.68	131.42	151.6	99.30	1117.3	2.29
25.2	4.78	132.21	148.6	100.11	1117.7	2.34
25.1	4.88	133.00	145.8	100.88	1118.0	2.39
25.0	4.98	133.77	143.0	101.65	1118.3	2.44
24.0	5.98	140.64	129.0	108.51	1121.3	2.93
23.0	6.98	146.78	104.5	114.64	1123.9	3.42
22.0	7.97	152.16	92.3	120.02	1126.2	3.90
21.0	8.97	157.00	82.6	124.86	1128.2	4.39
20.0	9.97	161.42	74.8	129.28	1130.1	4.88
19.0	10.97	165.42	68.5	133.28	1131.8	5.37
18.0	11.96	169.14	63.1	137.00	1133.4	5.86
17.0	12.96	172.63	58.6	140.50	1134.8	6.35
16.0	13.96	175.93	54.6	143.80	1136.1	6.84
15.0	14.95	179.03	51.17	146.91	1137.4	7.32
14.0	15.95	181.92	49.03	149.80	1138.6	7.81
13.0	16.95	184.68	45.55	152.57	1139.7	8.30
12.0	17.95	187.31	43.18	155.21	1140.7	8.79
11.0	18.94	189.83	41.05	157.73	1141.7	9.28
10.0	19.94	192.23	39.13	160.14	1142.3	9.77
9.0	20.94	194.52	37.40	162.44	1143.6	10.26
8.0	21.94	196.73	35.79	164.68	1144.5	10.75
7.0	22.93	198.87	34.33	166.81	1145.4	11.23
6.0	23.93	200.94	33.00	168.88	1146.3	11.72
5.0	24.93	202.92	31.76	170.89	1147.0	12.21
4.0	25.92	204.85	30.62	172.81	1147.6	12.70
3.0	26.92	206.71	29.55	174.68	1148.4	13.19
2.0	27.92	208.52	28.57	176.50	1149.1	13.68
1.0	28.92	210.28	27.66	178.27	1149.7	14.17
0.0	29.92	212.00	26.79	180.00	1150.4	14.67

Condenser Auxiliaries

With a jet condenser, there is provided an air pump for removing the air and condensation, and generally it is a direct acting steam pump, as shown in figure 79, page 194.

The injection water in condensing the steam in a jet condenser is pumped by the air pump, unless the source of supply is far distant, in which case a centrifugal or direct-acting pump is used for pumping the supply.

For surface condensers operating on 26 inches vacuum or less, a combined air and circulating pump is generally provided as shown in figure 84. The pump consists of a direct acting steam cylinder with an air cylinder on one end for removing air and condensate, and a circulating cylinder on the other end, which pumps cooling water through the condenser. The condenser may be mounted on the pump as shown, or on a separate foundation, and piped to the air and circulating pump.

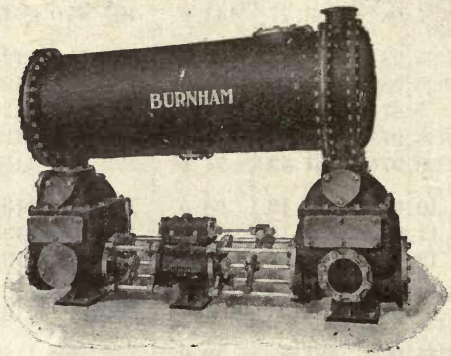


Fig. 84.
Surface Condenser Mounted on Air and Circulating Pump

Instead of a combined air and circulating pump, there may be provided separate pumps for handling the air and condensate. These may be either of the direct-acting steam-driven, or centrifugal types.

For 26″ of vacuum and lower, a dry-air pump is not necessary, provided a wet pump is used, which can handle both the air and condensate.

If a centrifugal type of condensate pump is used, a dry-air pump is necessary to remove the non-condensable vapors, as a centrifugal condensate pump can handle only the condensate.

For vacuums above 26″, surface condensers are generally provided with circulating pumps, which may be of the direct-acting steam-driven type, or the centrifugal circulating type.

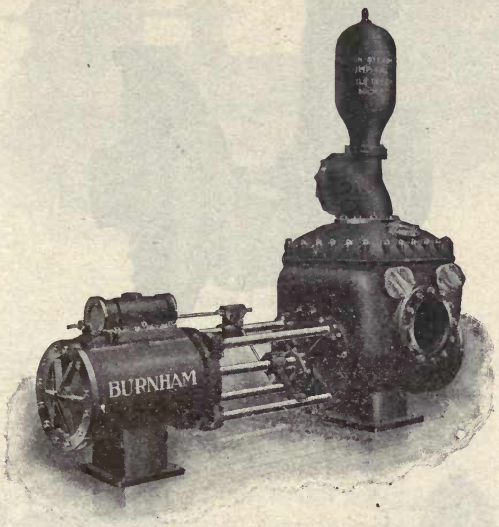


Fig: 85.
Burnham Direct Acting Steam Pump

The centrifugal circulating pump is especially suitable for this service on account of its simplicity, and saving in space and weight. It may be belt driven or operated by an electric motor, a steam turbine or a vertical steam-engine.

Circulating pumps generally operate on a low head of 20-30 feet, and this head consists of practically all friction head through the condenser.

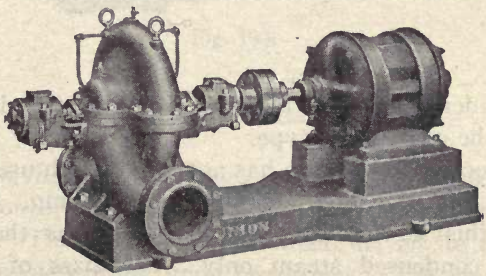


Fig 86.
Motor Driven Circulating Pump

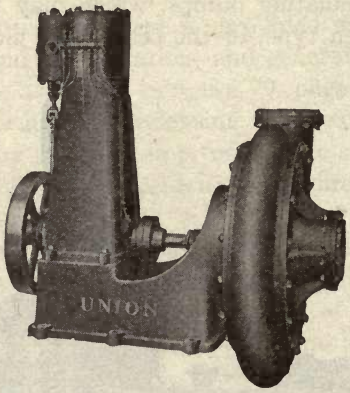


Fig. 87.
Steam Engine Driven Circulating Pump

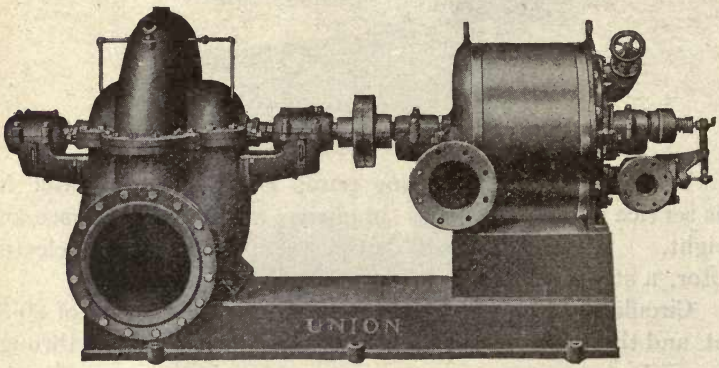


Fig. 88.
Steam Turbine Driven Circulating Pump

The condensate pump may be either of the direct-acting type, or of the centrifugal type.

The direct-acting pump has not been much used since the development of the centrifugal condensate pump. Centrifugal pumps for this service are generally small, as they have to handle the condensed steam only. The sizes of centrifugal condensate pumps generally run from 1½″ to 6″. A 6″ pump is sufficient for the largest units.

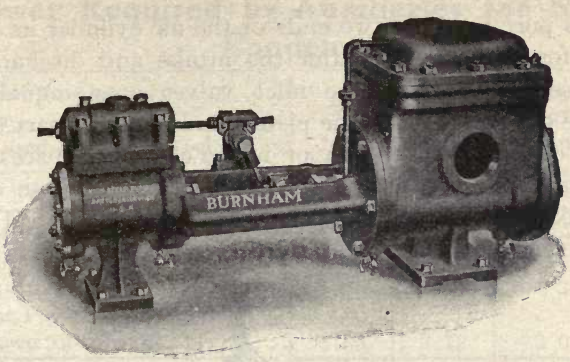


Fig. 89
Direct-Acting Condensate Pump

Centrifugal condensate pumps are built in the single stage types, either motor, or turbine driven. The head is rarely over 50 feet, of which approximately 30 feet is generally due to the vacuum in the condenser, and the remainder friction and delivery head in the discharge line. Centrifugal pumps should always be located below the condenser, so that there is a head of approximately 4 to 6 feet on the pump.

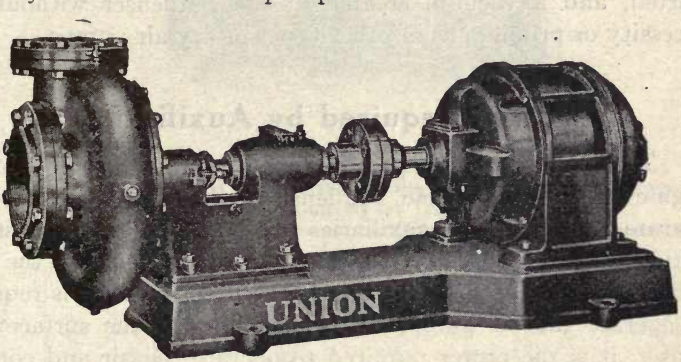


Fig. 193.

Centrifugal Condensate Pump

For removing the non-condensable vapors, a dry-air pump of the horizontal reciprocating type is provided as illustrated in figure 91. This pump may be steam driven, belt driven, or motor driven, depending upon conditions. The air pump is a displacement pump, and is fitted with simple flat disc air valves located in the ends of the cylinder, to reduce the clearance. A

by-pass port puts the two ends of the air cylinder in communication for a short time after the intake and discharge valves are closed and before the inlet valves open, equalizing the pressure on both sides of the piston, and avoiding part of the loss due to expansion of the vapor in the clearance spaces. Reciprocating air pumps are very economical in the use of power. The maximum power will be required at about 21″ of vacuum, and decreases as the vacuum increases, or improves.

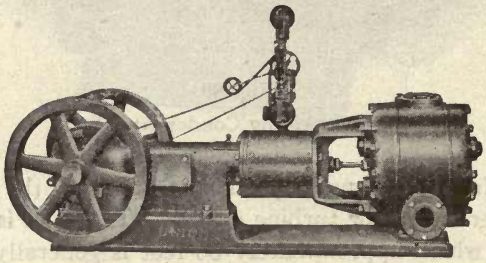


Fig 91
Steam Driven Dry Air Pump

With a reciprocating dry-air pump, the pump may be started, and a vacuum created in the condenser without the necessity or priming, as in other types of dry air pumps.

Power Required by Auxiliaries

To arrive at the net gain in economy by operating a steam engine or steam turbine condensing, the power required to operate the condenser auxiliaries must be charged against the initial saving in steam of the main unit produced by the condenser. With the jet condenser, this power is what is required to operate the air pump, and in the case of the surface condenser, it is the power required to operate the air and condensate pumps, as well as the circulating pump.

The power required by jet condensers varies from 1 H. P. per 1000 pounds of steam used per hour by the main engine, in the case of large units, to 3 H. P. per 1000 pounds of steam for small units.

The power required by the auxiliaries of a surface condenser varies from 2 to 5 per cent of the horse power of the main unit.

The following table gives the approximate power required for the surface condenser auxiliaries for a 300, 500, 1000 and 2000 K. W. turbine.

Power Required by Auxiliaries for Surface Condensers

K. W. on Turbine	300	500	1000	2000
Vacuum	28″	28″	28″	28″
Steam used by Turbine per Hr.	7000	10000	20000	30000
	Horse Power Used			
Circulating Pump	10	14	26	35
Dry Air Pump	4	6	12	18
Condensate Pump	.75	1	2	3
Per cent Power of Auxiliaries to Power of Turbine	3.65	3.12	3 0	2.1
Per cent Steam used by Auxiliaries to that used by Turbines	11.7	11.0	9.5	9.0

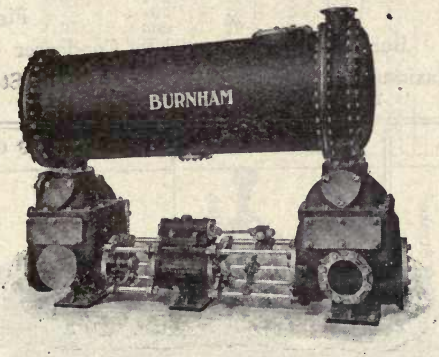


Fig. 84

Burnham Air and Circulating Pump, with Surface Condenser Mounted on Top

SIZE OF PUMP					Diameter of Pump Openings					
					For long pipe lines use larger pipes, reducing size at pump openings					
							Air Pump		Circulating Pump	
Diameter of Steam Cylinder	Diameter of Air Cylinder	Diameter of Circulating Water Cylinder	Length of Stroke	Gallons per Minute Circulating Cylinder will deliver at 100 feet per minute	Steam	Exhaust	Suction	Discharge	Suction	Discharge
7	8	8	10	261	1	4	4	4	6	5
8½	8	9	10	330	1½	4	4	6	6	5
8½	9	9	10	330	1	1½	4	4	6	5
8	10	10	12	408	1	1½	4	4	6	5
10	10	10	12	408	1¼	2	4	4	6	5
10	12	12	12	587	1¼	2	8	5	8	6
10	12	12	16	587	1¼	2	8	5	8	6
12	14	14	12	800	1½	2½	8	5	10	8
12	12	14	16	800	1½	2½	8	5	10	8
12	14	14	16	800	1½	2½	8	5	10	8
12	14	16	16	1040	1½	2½	8	6	10	8
14	16	16	16	1040	2	2½	10	6	10	8
14	16	16	20	1040	2	2½	10	6	12	10
14	18	18	20	1320	2	2½	12	8	12	10

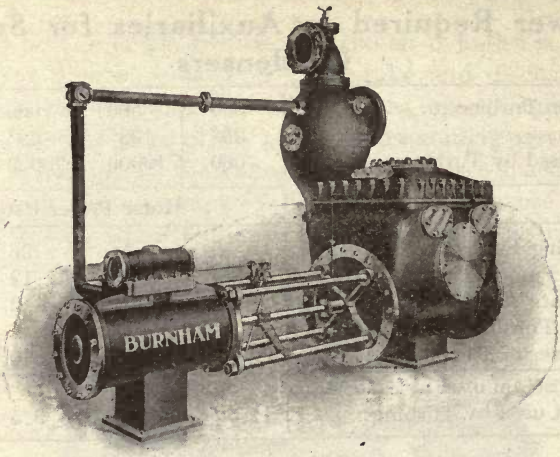


Fig. 79

Burnham Air Pump and Jet Condenser

Maximum Working Pressures: 200 Pounds Steam.
26″ Vacuum with 30″ Barometer

Size of Pump			CAPACITY		SIZE OF OPENINGS				
Diameter of Steam-Cylinder	Diameter of Air-Cylinder	Length of Stroke	Displacement in Gallons per Stroke	Pounds Steam Condensed per Hour—70° Cooling Water 26″ Vac. 30″ Bar.	Steam	Exhaust	Injection	Maximum Exhaust-Steam Inlet	Air-Pump Discharge
4½	5	8	.68	630	½	¾	2	4	2
4½	6	8	.98	910	½	¾	2	4	2½
5	6	10	1.22	1130	½	¾	2½	4	2½
5	7	10	1.66	1540	½	¾	2½	4	3
6⅛	8	10	2.17	2000	¾	1	3	8	3½
6⅛	9	10	2.75	2550	¾	1	4	8	3½
6⅛	10	10	3.40	3160	¾	1	4	8	4
6⅛	10	12	4.08	3800	¾	1	4	8	4
8½	10	10	3.40	3160	1	1½	4	8	4
8	10	12	4.08	3800	1	1½	4	8	4
8	12	12	5.87	5450	1	1½	6	12	5
10	12	12	5.87	5450	1¼	2	6	12	5
8	12	16	7.83	5450	1	1½	6	12	5
10	12	16	7.83	5450	1¼	2	6	12	5
8	14	16	10.66	7400	1	1½	6	12	6
10	14	16	10.66	7400	1¼	2	6	12	6
12	14	16	10.66	7400	1½	2½	6	12	6
10	16	16	13.92	9750	1¼	2	6	16	6
12	16	16	13.92	9750	1½	2½	6	16	6
10	16	20	17.40	9750	1½	2½	6	16	6
12	16	20	17.40	9750	1½	2½	6	16	6
12	18	20	22.02	12300	1½	2½	8	16	8
14	16	20	17.40	9750	2	2½	6	16	8
14	18	20	22.02	12300	2	2½	8	16	8
12	20	24	32.64	15000	1½	1½	8	16	8
14	20	24	32.64	15000	2	2½	8	16	8
12	22	24	39.49	18300	1½	2½	8	16	12
12	24	24	47.00	21700	1½	2½	10	20	12
14	22	24	39.49	18300	2	2½	8	16	12
14	24	24	47.00	21700	2	2½	10	20	12
14	26	24	55.16	25500	2	2½	10	24	12
14	28	24	63.97	30000	2	2½	10	24	14
16	26	24	55.16	25500	2	2½	10	24	12
16	28	24	63.97	30000	2	2½	10	24	14
16	30	24	73.43	34000	2	2½	10	24	16

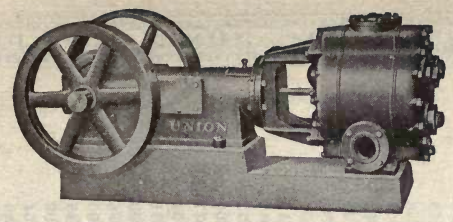


Fig. 93.

Details of Union Single Belt-Driven Enclosed-Type Dry-Vacuum Pump

Air Cylinder			Displacement Cu. Ft. Free Air			Pipe Openings			Driving Pulley		
Diameter	Stroke	Revolutions Per Minute	Per Revolution	Per Minute	Brake Horse Power Required at Peak Load	Suction	Discharge	Cooling Water Inlet	Diameter	Width of Belt	Diameter of Fly-Wheel
10	6	275	.545	150	6	3	2½	½	28	6	28
14	6	275	1.06	292	12	4	3½	½	28	6	28
18	6	275	1.76	483	18	5	5	½	28	6	28
18	8	250	2.35	590	22	5	5	½	42	8	42
22	8	250	3.52	880	31	7	6	½	42	8	42
22	10	235	4.40	1035	37	7	6	½	48	10	48
26	10	235	6.15	1445	49	8	7	½	48	10	48
28	12	220	8.56	1880	65	9	8	½	66	12	55
30	12	220	9.82	2160	75	10	8	½	66	12	55
32	15	210	13.98	2940	100	12	10	½	72	15	60

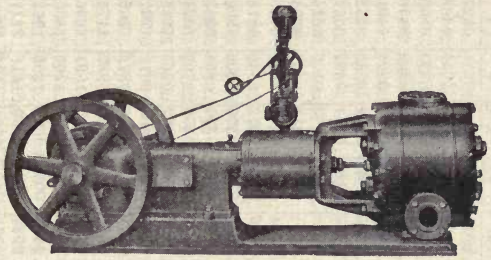


Fig. 91.

Details of Union Single Steam-Driven Enclosed-Type Dry-Vacuum Pump

Size of Pump				Displacem't Cu. Ft. Free Air				Pipe Openings					
Diameter Steam Cylinder	Diameter Air Cylinder	Stroke	Revolutions per Minute	Per Revolution	Per Minute	Required Steam Pressure	I. H. P. of Steam Cylinder at Peak Load	Steam	Exhaust	Suction	Discharge	Cooling Water Inlet	Diameter of Fly-Wheel
6⅛	10	6	275	.545	150	50	6	1½	2	3	2½	½	28
6⅛	14	6	275	1.06	292	60	12	1½	2	4	3½	½	28
6⅛	18	6	275	1.76	483	100	18	1½	2	5	5	½	28
8	14	6	275	1.06	292	35	12	2	2½	4	3½	½	28
8	18	6	275	1.76	483	50	18	2	2½	5	5	½	28
8	18	8	250	2.35	590	50	22	2	2½	5	5	½	36
8	22	8	250	3.52	874	90	31	2	2½	7	6	½	36
10	18	8	250	2.35	590	35	22	2½	3	5	5	½	36
10	22	8	250	3.52	874	50	31	2½	3	7	6	½	36
10	22	10	235	4.40	1035	70	37	3	3½	7	6	½	48
10	26	10	235	6.15	1432	85	49	3	3½	8	7	½	48
12	22	10	235	4.40	1035	35	37	3½	4	7	6	½	48
12	26	10	235	6.15	1432	50	49	3½	4	8	7	½	48
12	28	12	220	8.56	1880	80	65	3½	4	9	8	½	55
12	30	12	220	9.82	2160	100	75	3½	4	10	8	½	55
14	28	12	220	8.56	1880	40	65	4	5	9	8	½	55
14	30	12	220	9.82	2160	45	75	4	5	10	8	½	55
14	32	15	210	13.98	2940	80	100	4	5	12	10	½	60
18	32	15	210	13.98	2940	35	100	5	6	12	10	½	60

Surfaces of Tubes in Square Feet

Diameter in Inches	LENGTH IN INCHES																		
	1/8	1/4	3/8	1/2	5/8	3/4	7/8	1	2	3	4	5	6	7	8	9	10	11	12
1/8	.0014	.0027	.0041	.0055	.0068	.0082	.0095	.0109	.0218	.0327	.0436	.0545	.0654	.0764	.0873	.0982	.1091	.1200	.1309
9/16	.0015	.0031	.0046	.0061	.0077	.0092	.0107	.0123	.0245	.0368	.0491	.0614	.0736	.0859	.0982	.1104	.1227	.1350	.1473
5/8	.0017	.0034	.0051	.0068	.0085	.0102	.0119	.0136	.0273	.0409	.0545	.0682	.0818	.0954	.1091	.1227	.1364	.1500	.1636
11/16	.0019	.0037	.0056	.0075	.0094	.0112	.0131	.0150	.0300	.0450	.0600	.0750	.0900	.1050	.1200	.1350	.1500	.1650	.1800
3/4	.0020	.0041	.0061	.0082	.0102	.0123	.0143	.0164	.0327	.0491	.0654	.0818	.0982	.1145	.1309	.1473	.1636	.1800	.1963
13/16	.0022	.0044	.0066	.0089	.0111	.0133	.0155	.0177	.0355	.0532	.0709	.0886	.1064	.1241	.1418	.1595	.1773	.1950	.2127
7/8	.0024	.0048	.0072	.0095	.0119	.0143	.0167	.0191	.0382	.0573	.0764	.0954	.1154	.1336	.1527	.1718	.1909	.2100	.2291
15/16	.0026	.0051	.0077	.0102	.0128	.0153	.0179	.0205	.0409	.0614	.0818	.1023	.1227	.1432	.1636	.1841	.2045	.2250	.2454
1	.0027	.0055	.0082	.0109	.0136	.0164	.0191	.0218	.0436	.0654	.0873	.1091	.1309	.1527	.1745	.1963	.2182	.2400	.2618
1 1/8	.0031	.0061	.0092	.0123	.0153	.0184	.0215	.0245	.0491	.0736	.0982	.1227	.1473	.1718	.1963	.2209	.2454	.2700	.2945
1 1/4	.0034	.0068	.0102	.0136	.0170	.0205	.0239	.0273	.0545	.0818	.1091	.1364	.1636	.1909	.2182	.2454	.2727	.3000	.3272
1 3/8	.0037	.0075	.0112	.0150	.0187	.0225	.0262	.0300	.0600	.0900	.1200	.1500	.1800	.2100	.2400	.2700	.3000	.3300	.3600
1 1/2	.0041	.0082	.0123	.0164	.0205	.0245	.0286	.0327	.0654	.0982	.1309	.1636	.1963	.2291	.2618	.2945	.3272	.3600	.3927
1 5/8	.0044	.0089	.0133	.0177	.0222	.0266	.0310	.0355	.0709	.1064	.1418	.1773	.2127	.2482	.2836	.3191	.3545	.3900	.4254
1 3/4	.0048	.0095	.0143	.0191	.0239	.0286	.0334	.0382	.0764	.1145	.1527	.1909	.2291	.2673	.3054	.3436	.3818	.4200	.4581
1 7/8	.0051	.0102	.0153	.0205	.0256	.0307	.0358	.0409	.0818	.1227	.1636	.2045	.2454	.2863	.3272	.3682	.4091	.4500	.4909
2	.0055	.0109	.0164	.0218	.0273	.0327	.0382	.0436	.0873	.1309	.1745	.2182	.2618	.3054	.3491	.3927	.4363	.4800	.5236
2 1/4	.0061	.0123	.0184	.0245	.0307	.0368	.0430	.0491	.0982	.1473	.1963	.2454	.2945	.3436	.3927	.4418	.4909	.5400	.5890
2 1/2	.0068	.0136	.0205	.0273	.0341	.0409	.0477	.0545	.1091	.1636	.2182	.2727	.3272	.3818	.4363	.4909	.5454	.6000	.6545
2 3/4	.0075	.0150	.0225	.0300	.0375	.0450	.0525	.0600	.1200	.1800	.2400	.3000	.3600	.4200	.4800	.5400	.6000	.6600	.7199
3	.0082	.0164	.0245	.0327	.0409	.0491	.0573	.0654	.1309	.1963	.2618	.3272	.3927	.4581	.5236	.5890	.6545	.7199	.7854
3 1/4	.0089	.0177	.0266	.0355	.0443	.0532	.0620	.0709	.1418	.2127	.2836	.3545	.4254	.4963	.5672	.6381	.7090	.7799	.8508
3 1/2	.0095	.0191	.0286	.0382	.0477	.0573	.0668	.0764	.1527	.2291	.3054	.3818	.4581	.5345	.6109	.6872	.7636	.8399	.9163
3 3/4	.0102	.0205	.0307	.0409	.0511	.0614	.0716	.0818	.1636	.2454	.3222	.4091	.4909	.5727	.6545	.7363	.8181	.8999	.9817
4	.0109	.0218	.0327	.0436	.0545	.0654	.0764	.0873	.1745	.2618	.3491	.4363	.5236	.6109	.6981	.7854	.8727	.9599	1.047

Surfaces of Tubes in Square Feet

(*Continued*)

Diameter in Inches	LENGTH IN FEET													
	2	3	4	5	6	7	8	9	10	11	12	13	14	15
½	.2618	.3927	.5236	.6545	.7854	.9163	1.047	1.178	1.309	1.440	1.571	1.702	1.833	1.963
9/16	.2945	.4418	.5890	7363	.8836	1.031	1.178	1.325	1.473	1.620	1.767	1.914	2.062	2.209
⅝	.3272	.4909	.6545	.8181	.9817	1.145	1.309	1.473	1.636	1.800	1.963	2.127	2.291	2.454
11/16	.3600	.5400	.7199	.8999	1.080	1.260	1.440	1.620	1.800	1.980	2.160	2.340	2.520	2.700
¾	.3927	.5980	.7854	.9817	1.178	1.374	1.571	1.767	1.963	2.160	2.356	2.553	2.749	2.945
13/16	.4254	.6381	.8508	1.064	1.276	1.489	1.702	1.914	2.127	2.340	2.553	2.765	2.978	3.191
⅞	.4581	.6872	.9163	1.145	1.374	1.604	1.833	2.062	2.291	2.520	2.749	2.978	3.207	3.436
15/16	.4909	.7363	.9817	1.227	1.473	1.718	1.963	2.209	2.454	2.700	2.945	3.191	3.436	3.682
1	.5236	.7854	1.047	1.309	1.571	1.833	2.094	2.356	2.618	2.880	3.142	3.403	3.665	3.927
1⅛	.5890	.8836	1.178	1.473	1.767	2.062	2.356	2.651	2.945	3.240	3.534	3.829	4.123	4.418
1¼	.6545	.9817	1.309	1.636	1.963	2.291	2.618	2.945	3.272	3.600	3.927	4.254	4.581	4.909
1⅜	.7199	1.080	1.440	1.800	2.160	2.520	2.880	3.240	3.600	3.960	4.320	4.680	5.040	5.400
1½	.7854	1.178	1.571	1.963	2.356	2.749	3.142	3.534	3.927	4.320	4.712	5.105	5.498	5.890
1⅝	.8508	1.276	1.702	2.127	2.553	2.978	3.403	3.829	4.254	4.680	5.105	5.531	5.956	6.381
1¾	.9163	1.374	1.833	2.291	2.749	3.307	3.665	4.123	4.581	5.040	5.498	5.956	6.414	6.872
1⅞	.9817	1.473	1.963	2.454	2.945	3.436	3.927	4.418	4.909	5.400	5.890	6.381	6.872	7.363
2	1.047	1.571	2.094	2.618	3.142	3.665	4.189	4.712	5.236	5.760	6.283	6.807	7.330	7.854
2¼	1.178	1.767	2.356	2.945	3.534	4.123	4.712	5.301	5.890	6.480	7.069	7.658	8.247	8.836
2½	1.309	1.963	2.618	3.272	3.927	4.581	5.236	5.890	6.545	7.199	7.854	8.508	9.163	9.817
2¾	1.44	2.160	2.880	3.600	4.320	5.040	5.760	6.480	7.199	7.919	8.639	9.359	10.08	10.80
3	1.571	2.356	3.142	3.927	4.712	5.498	6.283	7.069	7.854	8.639	9.425	10.210	11.50	11.78
3¼	1.702	2.553	3.403	4.254	5.105	5.956	6.807	7.658	8.508	9.359	10.21	11.06	11.91	12.76
3½	1.833	2.749	3.665	4.581	5.498	6.414	7.330	8.247	9.163	10.08	11.00	11.91	12.83	13.74
3¾	1.963	2.945	3.927	4.909	5.890	6.872	7.854	8.836	9.817	10.80	11.78	12.76	13.74	14.73
4	2.094	3.142	4.189	5.236	6.283	7.330	8.378	9.425	10.47	11.52	12.57	13.61	14.66	15.71

Table Showing Weight Per Foot of Seamless Brass Tubes

Stub's or Birmingham Gauge, Measured in Outside Diameters

GAUGE NO.	3	4	5	6	7	8	9	10	11	12	13	14
Thickness of each No. in decimal parts of inch	.259	.238	.220	.203	.180	.165	.148	.134	.120	.109	.095	.083
Frac. of inch corresp'nding closely to Gauge Nos.:	1/4	15/64	.....	13/64	3/16	11/64	9/64	.....	1/8	.....	3/32	5/64
Diameter Tubes, In's.												
1/8	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
3/16	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
1/4	.....	.....	.....	.....	.....	.....	.....	.....	.18	.177	.170	.160
5/16	.....	.....	.....	.....	.....	.....	.....	.....	.27	.256	.238	.220
3/8	.....	.....	.....	.....	.....	.40	.39	.37	.35	.335	.307	.280
7/16	.....	.....	.....	.....	.....	.52	.49	.47	.44	.413	.376	.340
1/2	.....	.....	.....	.70	.66	.64	.60	.57	.53	.492	.444	.400
9/16	.....	.....	.....	.84	.79	.76	.71	.66	.61	.571	.513	.460
5/8	1.09	1.06	1.03	.99	.92	.88	.81	.76	.70	.649	.581	.520
11/16	1.28	1.23	1.19	1.13	1.05	.99	.92	.86	.79	.728	.650	.580
3/4	1.47	1.41	1.35	1.28	1.18	1.11	1.03	.95	.87	.807	.718	.640
13/16	1.65	1.58	1.50	1.43	1.31	1.23	1.13	1.05	.96	.885	.787	.700
7/8	1.84	1.75	1.66	1.57	1.44	1.35	1.24	1.15	1.04	.964	.855	.759
15/16	2.03	1.92	1.82	1.72	1.57	1.47	1.35	1.24	1.13	1.042	.924	.819
1	2.22	2.09	1.98	1.87	1.70	1.59	1.45	1.34	1.22	1.12	.99	.88
1 1/8	2.60	2.44	2.30	2.16	1.96	1.83	1.67	1.53	1.39	1.28	1.13	1.00
1 1/4	2.97	2.78	2.61	2.45	2.22	2.07	1.88	1.73	1.56	1.44	1.27	1.12
1 3/8	3.35	3.12	2.93	2.75	2.48	2.30	2.10	1.92	1.74	1.59	1.40	1.24
1 1/2	3.72	3.47	3.25	3.04	2.74	2.54	2.31	2.11	1.91	1.75	1.54	1.36
1 5/8	4.09	3.81	3.57	3.33	3.00	2.78	2.52	2.31	2.08	1.91	1.68	1.48
1 3/4	4.47	4.15	3.88	3.62	3.26	3.02	2.74	2.50	2.26	2 06	1.82	1.60
1 7/8	4.84	4.50	4.20	3.92	3.52	3.26	2.95	2.69	2.43	2.22	1.95	1.72
2	5.21	4.84	4.52	4.21	3.78	3.50	3.16	2.89	2.60	2.38	2.09	1.84
2 1/8	5.59	5.18	4.84	4.50	4.04	3.73	3.38	3.08	2.78	2.54	2.23	1.96
2 1/4	5.96	5.53	5.15	4.80	4.30	3.97	3.59	3.27	2.95	2.69	2.36	2.08
2 3/8	6.34	5.87	5.47	5.09	4.56	4.21	3.80	3.47	3.12	2.85	2.50	2.20
2 1/2	6.71	6.21	5.79	5.38	4.82	4.45	4.02	3.66	3.30	3.01	2.64	2.32
2 5/8	7.08	6.56	6.11	5.67	5.08	4.69	4.23	3.85	3.47	3.17	2.77	2.44
2 3/4	7.46	6.90	6.42	5.97	5.34	4.92	4.44	4.05	3.64	3.32	2.91	2.56
2 7/8	7.83	7.24	6.74	6.26	5.60	5.16	4.66	4.24	3.81	3.48	3.05	2.68
3	8.20	7.59	7.06	6.55	5.86	5.40	4.87	4.43	3.99	3.64	3.19	2.79
3 1/8	8.58	7.93	7.38	6.85	6.12	5.64	5.08	4.63	4.16	3.79	3.32	2.91
3 1/4	8.95	8.27	7.69	7.14	6.38	5.88	5.30	4.82	4.33	3.95	3.46	3.03
3 3/8	9.33	8.62	8.01	7.43	6.64	6.11	5.51	5.01	4.51	4.11	3.60	3.15
3 1/2	9.70	8.96	8.33	7.72	6.90	6.35	5.72	5.21	4.68	4.27	3.73	3.27
3 5/8	10.07	9.30	8.65	8.02	7.16	6.59	5.94	5.40	4.85	4.42	3.87	3.39
3 3/4	10.45	9.65	8.96	8.31	7.42	6.83	6.15	5.59	5.03	4.58	4.01	3.51
3 7/8	10.82	9.99	9.28	8.60	7.68	7.07	6.37	5.79	5.20	4.74	4.15	3.63

To determine weight per foot of a tube of a given Inside Diameter, add to weights in above list the weights given below under corresponding gauge numbers

GAUGE NO.	3	4	5	6	7	8	9	10	11	12	13	14
Increase in lbs. per foot	1.5487	1.3077	1.1174	.9514	.7480	.6285	.5057	.4145	.3324	.2743	.2084	.1590

Table Showing Weight per Foot of Seamless Brass Tubes

(Continued)

Stub's or Birmingham Gauge, Measured in Outside Diameters

GAUGE NO.	15	16	17	18	19	20	21	22	23	24	25	26	27
Thickness of each No. in decimal parts of inch:	.072	.065	.058	.049	.042	.035	.032	.028	.025	.022	.020	.018	.016
Frac. of inch correspondi'g closely to Gauge Nos.:	.....	1/16	.....	3/64	...	....	1/32	.....	.....	.....	.....	.....	1/64
Diameter Tubes, In's													
1/8	.....	.045	.045	.043	.040	.036	.034	.031	.029	.026	.024	.022	.020
9/64	.096	.092	.087	.078	.070	.062	.057	.051	.047	.042	.039	.035	.032
1/4	.148	.139	.129	.114	.101	.087	.080	.072	.065	.058	.053	.048	.043
5/32	.200	.186	.170	.149	.131	.112	.104	.092	.083	.074	.067	.061	.055
3/8	.252	.233	.212	.184	.161	.137	.127	.112	.101	.090	.082	.074	.066
7/32	.304	.279	.254	.220	.192	.163	.150	.132	119	.106	.096	.087	.078
1/2	.356	.326	.296	.255	.222	.188	.173	.152	.137	.121	.111	.100	.089
9/32	.408	.373	.338	.290	.252	.213	.196	.173	.155	.137	.125	.113	.101
5/8	.460	.420	.380	.326	.283	.238	.219	.193	.173	.153	.140	.126	.112
11/32	.511	.467	.421	.361	.313	.264	.242	.213	.191	.169	.154	.139	.124
3/4	.563	.514	.463	.396	.343	.289	.265	.233	.209	.185	.169	.152	.136
13/32	.615	.561	505	.432	.373	.314	.288	.253	.227	.201	.183	.165	.148
7/8	.667	.608	.547	.467	.404	.339	.311	.274	.245	.217	.197	.178	.159
15/32	.719	.655	.589	.502	.434	.365	.334	.294	.263	.232	.211	.191	.171
1	.77	.70	.63	.54	.46	.389	.358	.314	.281	.248	.226	.204	.182
1 1/8	.87	.79	.71	.61	.52	.439	.404	.354	.317	.280	.255	.230	.205
1 1/4	.98	.89	.80	.68	.59	.490	.450	.395	.354	.312	284	.256	.228
1 3/8	1.08	.98	.88	.75	.65	.540	.496	.435	.390	.343	.313	282	.251
1 1/2	1.19	1.08	.96	.82	.71	.591	.542	.476	.426	.375	.342	.308	.274
1 5/8	1.29	1.17	1.05	.89	.77	.641	.588	.516	.462	.407	.371	.334	.....
1 3/4	1.39	1.26	1.13	.96	.83	.692	.635	.556	.498	.439	.399	.360	.....
1 7/8	1.50	1.36	1.22	1.03	.89	.742	.681	.597	.534	.470	.428	.386	.....
2	1.60	1.45	1 30	1.10	.95	.793	.727	.637	.570	.502	.457	412	.....
2 1/8	1.71	1.55	1.38	1.17	1.01	.843	.773	.678	.606	.534	.486	.....	.....
2 1/4	1.81	1.64	1.47	1.24	1.07	.894	.819	.718	.642	.566	.515	.....	.....
2 3/8	1.91	1.73	1.55	1.32	1.13	.944	.866	.758	.678	.597	544	...	.....
2 1/2	2.02	1.83	1.63	1.39	1.19	.995	.912	.799	.714	.629	.573	.....	.....
2 5/8	2.12	1.92	1.72	1.46	1.25	1.045	.958	.839	.750	.661	.....	.....	.....
2 3/4	2.23	2.01	1.80	1.53	1.31	1.096	1.004	.880	.786	.693	.....	.....	.....
2 7/8	2.33	2.11	1.89	1.60	1.37	1.146	1.050	.920	.822	.724	.....	.....	.....
3	2.43	2.20	1.97	1.67	1.43	1.197	1.096	.960	.859	.756	.....	.....	.....
3 1/8	2.54	2.30	2.05	1.74	1.49	1.247	1.143	1.001	.895	.788	.....	.....	.....
3 1/4	2.64	2.39	2.14	1.81	1.55	1.298	1.189	1.041	.931	.820	.....	.....	.....
3 3/8	2.74	2.48	2.22	1.88	1.62	1.348	1.235	1.082	.967	.851	.....	.....	.....
3 1/2	2.85	2.58	2.30	1.95	1.68	1.399	1.281	1.122	1.003	.883	.....	.....	.....
3 5/8	2.95	2.67	2.39	2.02	1.74	1.449	1.327	1.162	1.039	.915	.....	.....	.....
3 3/4	3.06	2.76	2.47	2.09	1.80	1.50	1.373	1.203	1.075	.946	.....	.....	.....
3 7/8	3.16	2.86	2.56	2.16	1.86	1.55	1.42	1.243	1.111	.978	.....	.....	.....

To determine weight per foot of a tube of a given Inside Diameter, add to weights in above list the weights given below under corresponding gauge numbers.

GAUGE NO.	15	16	17	18	19	20	21	22	23	24	25	26	27
Increase in lbs. per foot:	.1197	.0975	.0777	.0554	.0407	.0283	.0236	.0181	.0144	.0112	.0092	.0075	.0059

Table Showing Weight Per Foot of Seamless Brass Tubes

(Continued)

Stub's or Birmingham Gauge, Measured in Outside Diameters

GAUGE NO.	3	4	5	6	7	8	9	10	11	12
Thickness of each No. in decimal parts of inch	.259	.238	.220	.203	.180	.165	.148	.134	.120	.109
Fraction of inch, corresponding closely to Gauge Nos.:	1/4	15/64	.....	13/64	3/16	11/64	9/64	.....	1/8″	.....
Diameter Tubes, Inches										
4	11.19	10.33	9.60	8.90	7.94	7.31	6.58	5.98	5.37	4.89
4 1/8.....	11.57	10.68	9.91	9.19	8.20	7.54	6.79	6.17	5.55	5.05
4 1/4.....	11.94	11.02	10.23	9.48	8.46	7.78	7.01	6.37	5.72	5.21
4 3/8.....	12.32	11.36	10.55	9.77	8.72	8.02	7.22	6.56	5.89	5.37
4 1/2.....	12.69	11.71	10.87	10.07	8.98	8.26	7.43	6.75	6.06	5.52
4 5/8.....	13.06	12.05	11.18	10.36	9.24	8.50	7.65	6.94	6.24	5.68
4 3/4.....	13.44	12.39	11.50	10.65	9.50	8.73	7.86	7.14	6.41	5.84
4 7/8.....	13.81	12.74	11.82	10.95	9.76	8.97	8.07	7.33	6.58	6.00
5	14.18	13.08	12.14	11.24	10.02	9.21	8.29	7.53	6.76	6.15
5 1/8.....	14.56	13.42	12.45	11.53	10.28	9.45	8.50	7.72	6.93	6.31
5 1/4.....	14.93	13.77	12.77	11.82	10.53	9.69	8.71	7.91	7.10	6.47
5 3/8.....	15.31	14.11	13.09	12.12	10.79	9.92	8.93	8.11	7.28	6.62
5 1/2.....	15.68	14.45	13.41	12.41	11.05	10.16	9.14	8.30	7.45	6.78
5 5/8.....	16.05	14.80	13.72	12.70	11.31	10.40	9.35	8.49	7.62	6.94
5 3/4.....	16.43	15.14	14.04	13.00	11.57	10.64	9.57	8.69	7.80	7.10
5 7/8.....	16.80	15.48	14.36	13.29	11.83	10.88	9.78	8.88	7.97	7.25
6	17.17	15.83	14.67	13.58	12.09	11.12	9.99	9.07	8.14	7.41
6 1/8.....	17.55	16.17	14.99	13.87	12.35	11.35	10.21	9.27	8.32	7.57
6 1/4.....	17.92	16.51	15.31	14.17	12.61	11.59	10.42	9.46	8.49	7.72
6 3/8.....	18.30	16.86	15.63	14.46	12.87	11.83	10.64	9.65	8.66	7.88
6 1/2.....	18.67	17.20	15.94	14.75	13.13	12.07	10.85	9.85	8.84	8.04
6 5/8.....	19.04	17.54	16.26	15.05	13.39	12.31	11.06	10.04	9.01	8.20
6 3/4.....	19.42	17.89	16.58	15.34	13.65	12.54	11.28	10.23	9.18	8.35
6 7/8.....	19.79	18.23	16.90	15.63	13.91	12.78	11.49	10.43	9.35	8.51
7	20.16	18.57	17.21	15.92	14.17	13.02	11.70	10.62	9.53	8.67
7 1/8.....	20.54	18.92	17.53	16.22	14.43	13.26	11.92	10.81	9.70	8.83
7 1/4.....	20.91	19.26	17.85	16.51	14.69	13.50	12.13	11.01	9.87	8.98
7 3/8.....	21.29	19.60	18.17	16.80	14.95	13.73	12.34	11.20	10.05	9.14
7 1/2.....	21.66	19.95	18.48	17.10	15.21	13.97	12.56	11.39	10.22	9.30
7 5/8.....	22.03	20.29	18.80	17.39	15.47	14.21	12.77	11.59	10.39	9.45
7 3/4.....	22.41	20.64	19.12	17.68	15.73	14.45	12.98	11.78	10.57	9.61
7 7/8.....	22.78	20.98	19.44	17.98	15.99	14.69	13.20	11.97	10.74	9.77
8	23.15	21.32	19.75	18.27	16.25	14.93	13.41	12.17	10.91	9.93

To determine weight per foot of a tube of a given Inside Diameter, add to weights in above list the weights given below under corresponding gauge numbers.

GAUGE NO.	3	4	5	6	7	8	9	10	11	12
Increase in Pounds Per Foot	1.5487	1.3077	1.1174	.9514	.7480	.6285	.5057	.4145	.3324	.2743

Table Showing Weight per Foot of Seamless Brass Tubes

(Continued)

Stub's or Birmingham Gauge, Measured in Outside Diameters

GAUGE NO.	13	14	15	16	17	18	19	20	21	22	23	24
Thickness of each No. in decimal parts of inch:	.095	.083	.072	.065	.058	.049	.042	.035	.032	.028	.025	.022
Frac. of inch correspondi'g closely to Gauge Nos.:	3/32	5/64	.....	1/16	.....	3/64	.....	.....	1/32	.....	.....	.....
Diameter Tubes, In's												
4	4.28	3.75	3.26	2.95	2.6.	2.23	1.92	1.601	1.466	1.284	1.147	1.010
4⅛....	4.42	3.87	3.37	3.05	2.72	2.30	1.98	1.651	1.512	1.324	1.183	.....
4¼....	4.56	3.99	3.47	3.14	2.81	2.38	2.04	1.702	1.558	1.364	1.219	.....
4⅜....	4.69	4.11	3.58	3.23	2.89	2.45	2.10	1.752	1.604	1.405	1.255	.....
4½....	4.8.	4.23	3.68	3.33	2.97	2.52	2.16	1.803	1.650	1.445	1.291	.....
4⅝....	4.97	4.35	3.78	3.42	3.06	2.59	2.22	1.853	1.697	1.486	.....	.....
4¾....	5.11	4.47	3.89	3.52	3.14	2.66	2.28	1.904	1.743	1.526	.....	.....
4⅞....	5.24	4.59	3.99	3.61	3.22	2.73	2.34	1.954	1.789	1.566	.....	.....
5	5.38	4.71	4.09	3.70	3.31	2.80	2.40	2.005	1.835	1.607	.....	.....
5⅛....	5.52	4.83	4.20	3.79	3.39	2.87	2.46	2.055	1.881	.....	.....	.....
5¼....	5.65	4.95	4.30	3.89	3.48	2.94	2.52	2.106	1.928	.....	.....	.....
5⅜....	5.79	5.07	4.41	3.98	3.56	3.01	2.58	2.156	1.974	.....	.....	.....
5½....	5.93	5.19	4.51	4.08	3.64	3.08	2.65	2.207	2.02	.....	.....	.....
5⅝....	6.07	5.31	4.61	4.17	3.73	3.15	2.71	2.257	.....	.....	.....	.....
5¾....	6.20	5.43	4.72	4.26	3.81	3.22	2.77	2.308	.....	.....	.....	.....
5⅞....	6.34	5.55	4.82	4.36	3.89	3.29	2.83	2.358	.....	.....	.....	.....
6	6.48	5.67	4.93	4.45	3.98	3.37	2.89	2.409	.....	.....	.....	.....
6⅛....	6.61	5.79	5.03	4.54	4.06	3.44	.....	.....	.....	.....	.....	.....
6¼....	6.75	5.91	5.13	4.64	4.15	3.51	.....	.....	.....	.....	.....	.....
6⅜....	6.89	6.03	5.24	4.73	4.2.	3.58	.....	.....	.....	.....	.....	.....
6½....	7.03	6.15	5.34	4.8.	4.31	3.65	.....	.....	.....	.....	.....	.....
6⅝....	7.16	6.27	5.45	4.92	4.40	3.72	.....	.....	.....	.....	.....	.....
6¾....	7.30	6.39	5.55	5.01	4.48	3.79	.....	.....	.....	.....	.....	.....
6⅞....	7.44	6.51	5.65	5.11	4.56	3.86	.....	.....	.....	.....	.....	.....
7	7.57	6.63	5.76	5.2.	4.6.	3.93	.....	.....	.....	.....	.....	.....
7⅛....	7.71	6.75	5.86	5.29	.....	.....	.....	.....	.....	.....	.....	.....
7¼....	7.85	6.87	5.96	5.39	.....	.....	.....	.....	.....	.....	.....	.....
7⅜....	7.99	6.99	6.07	5.48	.....	.....	.....	.....	.....	.....	.....	.....
7½....	8.12	7.11	6.17	5.58	.....	.....	.....	.....	.....	.....	.....	.....
7⅝....	8.26	7.23	6.28	5.67	.....	.....	.....	.....	.....	.....	.....	.....
7¾....	8.40	7.35	6.38	5.76	.....	.....	.....	.....	.....	.....	.....	.....
7⅞....	8.53	7.47	6.48	5.86	.....	.....	.....	.....	.....	.....	.....	.....
8	8.67	7.58	6.59	5.95	.....	.....	.....	.....	.....	.....	.....	.....

To determine weight per foot of a tube of a given Inside Diameter, add to weights in above list the weights given below under corresponding gauge numbers.

GAUGE NO.	13	14	15	16	17	18	19	20	21	22	23	24
Increase in Lbs. per Foot	.2084	.1590	.1197	.0975	.0777	.0554	.0407	.0283	0236	.0181	.0144	.0112

Table Showing Weight Per Foot of Seamless Brass Tubes

American or B. & S. Gauge, Measured in Outside Diameters

GAUGE NO.	2	3	4	5	6	7	8	9	10	11	12	13
Thickness of each No. in decimal parts of inch	.25763	.22942	.20431	.18194	.16202	.14428	.12849	.11443	.10189	.090742	.080808	.071961
Frac. of inch correspondi'g closely to Gauge Nos.:	1/4	15/64	13/64	3/16	11/64	9/64	1/8	7/64	.....	3/32	5/64	.....
Diameter Tubes, In's												
1/8	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
3/16	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
1/4	.....	.....	.....	.....	.....	.....	.....	.....	.174	.167	.16	.15
5/16	.....	.....	.....	.....	.....	.....	.....	.....	.25	.23	.22	.20
3/8	.....	.....	.....	.....	.....	.38	.36	.34	.32	.30	.27	.25
7/16	.....	.....	.....	.....	.....	.49	.46	.43	.39	.36	.33	.31
1/2	.....	.....	.....	.67	.63	.59	.55	.51	.47	.43	.39	.36
9/16	.....	.....	.....	.80	.75	.70	.64	.59	.54	.49	.45	.41
5/8	1.09	1.05	.99	.93	.87	.80	.74	.67	.61	.56	.51	.46
11/16	1.28	1.21	1.14	1.06	.98	.90	.83	.76	.69	.63	.57	.51
3/4	1.46	1.38	1.29	1.19	1.10	1.01	.92	.84	.76	.69	.62	.56
13/16	1.65	1.55	1.43	1.32	1.22	1.11	1.01	.92	.83	.75	.68	.61
7/8	1.84	1.71	1.58	1.45	1.33	1.22	1.11	1.00	.91	.82	.74	.67
15/16	2.02	1.87	1.73	1.59	1.45	1.32	1.20	1.09	.98	.89	.80	.72
1.....	2.21	2.04	1.88	1.71	1.57	1.42	1.29	1.17	1.06	.95	.86	.77
1 1/16....	2.58	2.37	2.17	1.98	1.80	1.63	1.48	1.33	1.20	1.08	.97	.87
1 1/4....	2.95	2.70	2.47	2.24	2.03	1.84	1.66	1.50	1.35	1.21	1.09	.98
1 3/8....	3.32	3.03	2.76	2.50	2.27	2.05	1.85	1.66	1.50	1.34	1.21	1.08
1 1/2....	3.69	3.36	3.05	2.77	2.50	2.26	2.03	1.83	1.64	1.47	1.32	1.19
1 5/8....	4.07	3.69	3.35	3.03	2.74	2.46	2.22	1.99	1.79	1.61	1.44	1.29
1 3/4....	4.44	4.03	3.64	3.29	2.97	2.67	2.40	2.16	1.94	1.74	1.56	1.39
1 7/8....	4.81	4.36	3.94	3.55	3.20	2.88	2.59	2.33	2.08	1.87	1.67	1.50
2	5.18	4.69	4.23	3.82	3.44	3.09	2.77	2.49	2.23	2.00	1.79	1.60
2 1/8....	5.55	5.02	4.53	4.08	3.67	3.30	2.96	2.66	2.38	2.13	1.91	1.71
2 1/4....	5.92	5.35	4.82	4.34	3.90	3.51	3.15	2.82	2.53	2.26	2.02	1.81
2 3/8....	6.30	5.68	5.12	4.60	4.14	3.71	3.33	2.99	2.67	2.39	2.14	1.91
2 1/2....	6.67	6.01	5.41	4.87	4.37	3.92	3.52	3.15	2.82	2.52	2.26	2.02
2 5/8....	7.04	6.34	5.71	5.13	4.61	4.13	3.70	3.32	2.97	2.65	2.37	2.12
2 3/4....	7.41	6.67	6.00	5.39	4.84	4.34	3.89	3.48	3.11	2.78	2.49	2.22
2 7/8....	7.78	7.00	6.30	5.65	5.07	4.55	4.07	3.65	3.26	2.91	2.61	2.33
3	8.16	7.34	6.59	5.92	5.31	4.75	4.26	3.81	3.41	3.05	2.72	2.43
3 1/8....	8.53	7.67	6.89	6.18	5.54	4.96	4.44	3.98	3.55	3.18	2.84	2.54
3 1/4....	8.90	8.00	7.18	6.44	5.77	5.17	4.63	4.14	3.70	3.31	2.96	2.64
3 3/8....	9.27	8.33	7.48	6.70	6.01	5.38	4.81	4.31	3.85	3.44	3.07	2.74
3 1/2....	9.64	8.66	7.77	6.97	6.24	5.59	5.00	4.47	4.00	3.57	3.19	2.85
3 5/8....	10.01	8.99	8.07	7.23	6.48	5.79	5.18	4.64	4.14	3.70	3.31	2.95
3 3/4....	10.39	9.32	8.36	7.49	6.71	6.00	5.37	4.80	4.29	3.83	3.42	3.06
3 7/8....	10.76	9.65	8.65	7.75	6.94	6.21	5.55	4.97	4.44	3.96	3.54	3.16

To determine weight per foot of a tube of a given Inside Diameter, add to weights in above list the weights given below under corresponding gauge numbers.

GAUGE NO.	2	3	4	5	6	7	8	9	10	11	12	13
Increase in lbs. per foot:	1.532	1.213	.9637	.7642	.6061	.4806	.3811	.3023	.2397	.1901	.1507	.1195

Table Showing Weight per Foot of Seamless Brass Tubes

(Continued)

American or B. & S. Gauge, Measured in Outside Diameters

GAUGE NO.	14	15	16	17	18	19	20	21	22	23	24	25	26
Thickness of each No. in decimal parts of inch:	.064084	.057068	.05082	.045257	.040303	.03589	.031961	.028462	.025347	.022571	.0201	.0179	.01594
Frac. of inch correspondi'g closely to Gauge Nos.	1/16	.....	.....	3/64	.....	.....	1/32	.....	.....	.....	.....	.....	1/64
Diameter Tubes, In's													
1/8	.....	.045	.043	.041	.039	.037	.034	.032	.028	.027	.024	.022	.020
3/16	.090	.086	.08	.07	.068	.062	.057	.053	.047	.043	.038	.035	.032
1/4	.14	.13	.12	.11	.097	.088	.080	.073	.065	.059	.053	.048	.043
5/16	.18	.17	.15	.14	.13	.114	.104	.094	.084	.076	.067	.061	.054
3/8	.23	.21	.19	.17	.15	.14	.126	.114	.102	.092	.082	.074	.066
7/16	.28	.25	.23	.20	.18	.17	.15	.135	.121	.108	.096	.087	.077
1/2	.32	.29	.26	.24	.21	.19	.17	.155	.139	.124	.111	.100	.089
9/16	.37	.33	.30	.27	.24	.22	.20	.176	.156	.141	.125	.113	.100
5/8	.42	.37	.34	.30	.27	.24	.22	.196	.174	.157	.140	.126	.112
11/16	.46	.42	.37	.33	.30	.27	.24	.22	.193	.173	.154	.139	.123
3/4	.51	.46	.41	.37	.33	.30	.26	.24	.211	.189	.169	.152	.135
13/16	.55	.50	.45	.40	.36	.32	.29	.26	.230	.206	.183	.164	.146
7/8	.60	.54	.48	.43	.39	.35	.31	.28	.248	.222	.198	.177	.158
15/16	.64	.58	.52	.47	.42	.37	.33	.30	.267	.238	.212	.190	.169
1	.69	.62	.56	.50	.45	.40	.36	.32	.285	.254	.227	.203	.181
1 1/8	.79	.70	.63	.57	.50	.45	.40	.36	.321	.297	.256	.229	.....
1 1/4	.88	.79	.70	.63	.56	.50	.45	.40	.358	.320	.285	.255	.....
1 3/8	.97	.87	.78	.69	.62	.55	.50	.44	.395	.352	.314	.281	.....
1 1/2	1.06	.95	.85	.76	.68	.61	.54	.48	.43	.384	.343	.317	....
1 5/8	1.16	1.03	.92	.82	.74	.66	.59	.52	.47	.417	.372	.....	.....
1 3/4	1.25	1.12	1.00	.89	.79	.71	.63	.56	.50	.450	.401	.....	.....
1 7/8	1.34	1.20	1.07	.95	.85	.76	.68	.61	.54	.482	.430	.....	.....
2	1.43	1.28	1.14	1.02	.91	.81	.73	.65	.58	.514	.459	.....	.....
2 1/8	1.53	1.36	1.22	1.09	.97	.86	.77	.69	.61	.558	.....	.....	.....
2 1/4	1.62	1.44	1.29	1.16	1.03	.92	.82	.73	.65	.580	.....	.....	.....
2 3/8	1.71	1.53	1.36	1.22	1.08	.97	.86	.77	.69	.612	.....	.....	.....
2 1/2	1.80	1.61	1.44	1.28	1.14	1.02	.91	.81	.73	.644	.....	.....	.....
2 5/8	1.90	1.69	1.51	1.35	1.20	1.07	.96	.85	.76	.....	.....	.....	.....
2 3/4	1.99	1.77	1.58	1.41	1.26	1.12	1.00	.89	.80	.....	.....	.....	.....
2 7/8	2.08	1.86	1.66	1.48	1.32	1.17	1.05	.93	.83	.....	.....	.....	...
3	2.17	1.94	1.73	1.54	1.38	1.23	1.09	.97	.87	.....	.....	.....	.....
3 1/8	2.27	2.02	1.80	1.62	1.43	1.28	1.14	1.02	.91	.....	.....	.....	.....
3 1/4	2.36	2.10	1.88	1.68	1.49	1.33	1.19	1.06	.94	.....	.....	.....	.....
3 3/8	2.45	2.19	1.95	1.74	1.55	1.38	1.23	1.10	.98	.....	.....	.....	.....
3 1/2	2.54	2.27	2.02	1.80	1.61	1.43	1.28	1.14	1.02	.....	.....	.....	.....
3 5/8	2.64	2.35	2.10	1.87	1.67	1.49	1.33	1.18	1.05	.....	.....	.....	.....
3 3/4	2.73	2.43	2.17	1.93	1.72	1.54	1.37	1.22	1.09	.....	.....	.....	.....
3 7/8	2.82	2.52	2.24	2.00	1.78	1.59	1.42	1.26	1.13	.....	.....	.....	.....

To determine weight per foot of a tube of a given Inside Diameter, add to weights in above list the weights given below in corresponding gauge numbers.

GAUGE NO.	14	15	16	17	18	19	20	21	22	23	24	25	26
Increase in lbs. per foot	.0948	.0752	.0596	.0473	.0375	.0297	.0236	.0187	.0148	.0117	.009	.0074	.0059

AND CONDENSERS FOR EVERY SERVICE

Table Showing Weight Per Foot of Seamless Brass Tubes

(Continued)

American or B. & S. Gauge, Measured in Outside Diameters

GAUGE NO.	2	3	4	5	6	7	8	9	10	11
Thickness of each No. in decimal parts of inch	.25763	.22942	.20431	.18194	.16202	.14428	.12849	.11443	.10189	.090742
Frac. of inch corresponding closely to Gauge No.:	1/4	15/64	13/64	3/16	11/64	9/64	1/8	7/64	.....	3/32
Diameter Tubes, Inches										
4	11.13	9.98	8.95	8.02	7.18	6.42	5.74	5.15	4.58	4.09
4⅛.....	11.50	10.31	9.24	8.28	7.41	6.63	5.93	5.30	4.73	4.22
4¼.....	11.87	10.65	9.54	8.54	7.64	6.84	6.11	5.46	4.88	4.35
4⅜.....	12.24	10.98	9.83	8.80	7.88	7.04	6.30	5.63	5.02	4.49
4½.....	12.62	11.31	10.13	9.07	8.11	7.25	6.48	5.79	5.17	4.62
4⅝.....	12.99	11.64	10.42	9.33	8.35	7.46	6.67	5.96	5.32	4.75
4¾.....	13.36	11.97	10.72	9.59	8.58	7.67	6.85	6.12	5.47	4.88
4⅞.....	13.73	12.30	11.01	9.85	8.81	7.88	7.04	6.29	5.61	5.01
5	14.10	12.63	11.31	10.12	9.05	8.08	7.22	6.45	5.76	5.14
5⅛.....	14.47	12.96	11.60	10.38	9.28	8.29	7.41	6.62	5.91	5.27
5¼.....	14.85	13.29	11.90	10.64	9.51	8.50	7.59	6.78	6.05	5.40
5⅜.....	15.22	13.62	12.19	10.90	9.75	8.71	7.78	6.95	6.20	5.53
5½.....	15.59	13.96	12.49	11.17	9.98	8.92	7.97	7.11	6.35	5.66
5⅝.....	15.96	14.29	12.78	11.43	10.22	9.12	8.15	7.28	6.49	5.79
5¾.....	16.33	14.62	13.08	11.69	10.45	9.33	8.34	7.44	6.64	5.92
5⅞.....	16.71	14.95	13.37	11.95	10.68	9.54	8.52	7.61	6.79	6.06
6	17.08	15.28	13.67	12.22	10.92	9.75	8.71	7.77	6.94	6.19
6⅛.....	17.45	15.61	13.96	12.48	11.15	9.96	8.89	7.94	7.08	6.32
6¼.....	17.82	15.94	14.26	12.74	11.38	10.17	9.08	8.10	7.23	6.45
6⅜.....	18.19	16.27	14.55	13.00	11.62	10.37	9.26	8.27	7.38	6.58
6½.....	18.56	16.60	14.84	13.27	11.85	10.58	9.45	8.43	7.52	6.71
6⅝.....	18.94	16.93	15.14	13.55	12.09	10.79	9.63	8.60	7.67	6.84
6¾.....	19.31	17.27	15.43	13.79	12.32	11.00	9.82	8.77	7.82	6.97
6⅞.....	19.68	17.60	15.73	14.05	12.55	11.21	10.00	8.93	7.96	7.10
7	20.05	17.93	16.02	14.32	12.79	11.41	10.19	9.10	8.11	7.23
7⅛.....	20.42	18.26	16.32	14.58	13.02	11.62	10.38	9.26	8.26	7.36
7¼.....	20.79	18.59	16.61	14.84	13.25	11.83	10.56	9.43	8.41	7.50
7⅜.....	21.17	18.92	16.91	15.10	13.49	12.04	10.75	9.59	8.55	7.63
7½.....	21.54	19.25	17.20	15.37	13.72	12.25	10.93	9.76	8 70	7.76
7⅝.....	21.91	19.58	17.50	15.63	13.96	12.45	11.12	9.92	8.85	7.89
7¾.....	22.28	19.91	17.79	15.89	14.19	12.66	11.30	10.09	8.99	8.02
7⅞.....	22.65	20.24	18.09	16.15	14.42	12.87	11.49	10.25	9.14	8.15
8	23.03	20.58	18.38	16.42	14.66	13.08	11.67	10.42	9.29	8.28

To determine weight per foot of a tube of a given Inside Diameter, add to weights in the above list the weights given below under corresponding gauge numbers.

GAUGE NO.	2	3	4	5	6	7	8	9	10	11
Increase in lbs. per foot	1.532	1.213	.9637	.7642	.6061	.4806	.3811	.3023	.2397	.1901

Table Showing Weight Per Foot of Seamless Brass Tubes

(Continued)

American or B. & S. Gauge, Measured in Outside Diameters

GAUGE NO.	12	13	14	15	16	17	18	19	20	21	22	23
Thickness of each No. in decimal parts of inch	.080808	.071961	.064084	.057068	.05082	.045257	.040303	.03589	.031961	.028462	.025347	.022571
Frac. of inch correspondi'g closely to Gauge Nos.	5/64	.....	1/16	.....	.....	3/64	.....	.....	1/32	.....	.....	.....
Diameter Tubes, In's												
4	3.66	3.26	2.91	2.60	2.32	2.06	1.84	1.64	1.46	1.30	1.16	.....
4⅛....	3.77	3.37	3.01	2.68	2.39	2.14	1.90	1.69	1.51	1.34	.....	.....
4¼....	3.89	3.47	3.10	2.76	2.46	2.20	1.96	1.74	1.55	1.39	.....	.....
4⅜....	4.01	3.58	3.19	2.84	2.54	2.26	2.01	1.80	1.60	1.43	.....	.....
4½....	4.12	3.68	3.28	2.93	2.61	2.32	2.07	1.85	1.64	1.47	.....	.....
4⅝....	4.24	3.78	3.38	3.01	2.68	2.39	2.13	1.90	1.69	.....	.....	.....
4¾....	4.36	3.89	3.47	3.09	2.76	2.46	2.19	1.95	1.74	.....	.....	.....
4⅞....	4.47	3.99	3.56	3.17	2.83	2.52	2.25	2.00	1.79	.....	.....	.....
5	4.59	4.09	3.65	3.26	2.90	2.59	2.31	2.05	1.83	.....	.....	.....
5⅛....	4.71	4.20	3.75	3.34	2.98	2.66	2.36	2.11	.....	.....	.....	.....
5¼....	4.82	4.30	3.84	3.42	3.05	2.72	2.42	2.16	.....	.....	.....	.....
5⅜....	4.94	4.41	3.93	3.50	3.12	2.78	2.48	2.21	.....	.....	.....	.....
5½....	5.06	4.51	4.02	3.59	3.20	2.85	2.54	2.26	.....	.....	.....	.....
5⅝....	5.17	4.61	4.12	3.67	3.27	2.91	2.60	.....	.....	.....	.....	.....
5¾....	5.29	4.72	4.21	3.75	3.34	2.98	2.65	.....	.....	.....	.....	.....
5⅞....	5.41	4.82	4.30	3.83	3.42	3.04	2.71	.....	.....	.....	.....	.....
6	5.52	4.93	4.39	3.92	3.49	3.11	2.77	.....	.....	.....	.....	.....
6⅛....	5.64	5.03	4.49	4.00	3.57	.....	.....	.....	.....	.....	.....	.....
6¼....	5.76	5.13	4.58	4.08	3.64	.....	.....	.....	.....	.....	.....	.....
6⅜....	5.87	5.24	4.67	4.16	3.71	.....	.....	.....	.....	.....	.....	.....
6½....	5.99	5.34	4.76	4.25	3.78	.....	.....	.....	.....	.....	.....	.....
6⅝....	6.11	5.45	4.86	4.33	3.85	.....	.....	.....	.....	.....	.....	.....
6¾....	6.22	5.55	4.95	4.41	3.93	.....	.....	.....	.....	.....	.....	.....
6⅞....	6.34	5.65	5.04	4.49	4.01	.....	.....	.....	.....	.....	.....	.....
7	6.46	5.76	5.13	4.57	4.08	.....	.....	.....	.....	.....	.....	.....
7⅛....	6.57	5.86	5.23	.....	.....	.....	.....	.....	.....	.....	.....	.....
7¼....	6.69	5.96	5.32	.....	.....	.....	.....	.....	.....	.....	.....	.....
7⅜....	6.80	6.07	5.41	.....	.....	.....	.....	.....	.....	.....	.....	.....
7½....	6.92	6.17	5.50	.....	.....	.....	.....	.....	.....	.....	.....	.....
7⅝....	7.04	6.28	5.60	.....	.....	.....	.....	.....	.....	.....	.....	.....
7¾....	7.15	6.38	5.69	.....	.....	.....	.....	.....	.....	.....	.....	.....
7⅞....	7.27	6.48	5.78	.....	.....	.....	.....	.....	.....	.....	.....	.....
8	7.39	6.59	5.87	.....	.....	.....	.....	.....	.....	.....	.....	.....

To determine weight per foot of a tube ot a given Inside Diameter, add to weights in the above list the weights given below under corresponding gauge numbers.

GAUGE NO.	12	13	14	15	16	17	18	19	20	21	22	23
Increase in lbs. per foot:	.1507	.1195	.0948	.0752	.0596	.0473	.0375	.0297	.0236	.0187	.0148	.0117

Hyperbolic Logarithms

No.	Log.	No.	Log.	No	Log.	No.	Log.	No.	Log.	No.	Log.
1.01	.0099	2.30	.8329	3.68	1.3029	5.06	1.6214	6.44	1.8625	7.82	2.0567
1.02	.0198	2.32	.8416	3.70	1.3083	5.08	1.6253	6.46	1.8656	7.84	2.0592
1.03	.0296	2.34	.8502	3.72	1.3137	5.10	1.6292	6.48	1.8687	7.86	2.0618
1.04	.0392	2.36	.8587	3.74	1.3191	5.12	1.6332	6.50	1.8718	7.88	2.0643
1.05	.0488	2.38	.8671	3.76	1.3244	5.14	1.6371	6.52	1.8749	7.90	2.0669
1.06	.0583	2.40	.8755	3.78	1.3297	5.16	1.6409	6.54	1.8779	7.92	2.0694
1.07	.0677	2.42	.8833	3.80	1.3350	5.18	1.6448	6.56	1.8810	7.94	2.0719
1.08	.0770	2.44	.8920	3.82	1.3403	5.20	1.6487	6.58	1.8840	7.96	2.0744
1.09	.0862	2.46	.9002	3.84	1.3455	5.22	1.6525	6.60	1.8871	7.98	2.0769
1.10	.0953	2.48	.9083	3.86	1.3507	5.24	1.6563	6.62	1.8901	8.00	2.0794
1.12	.1133	2.50	9163	3.88	1.3558	5.26	1.6601	6.64	1.8931	8.02	2.0819
1.14	.1310	2 52	.9243	3.90	1.3610	5.28	1.6639	6.66	1.8961	8.04	2.0844
1.16	.1484	2.54	.9322	3.92	1.3661	5.30	1.6677	6.68	1.8991	8.06	2.0869
1.18	.1655	2.56	.9400	3.94	1.3712	5.32	1.6715	6.70	1.9021	8.08	2.0894
1.20	.1823	2.58	9478	3.96	1.3762	5 34	1.6752	6.72	1.9051	8.10	2.0919
1.22	.1988	2.60	.9555	3.98	1.3813	5.36	1.6790	6.74	1.9081	8.12	2.0943
1.24	.2151	2 62	.9632	4.00	1.3863	5.38	1.6827	6.76	1.9110	8.14	2.0968
1.26	.2311	2.64	.9708	4.02	1.3913	5.40	1.6864	6.78	1.9140	8.16	2.0992
1.28	.2469	2.66	.9783	4.04	1.3962	5.42	1.6901	6.80	1.9169	8.18	2.1017
1.30	.2624	2.68	.9858	4.06	1.4012	5.44	1.6938	6.82	1.9199	8.20	2.1041
1.32	.2776	2.70	9933	4.08	1.4061	5.46	1.6974	6.84	1.9228	8.22	2.1066
1.34	.2927	2.72	1.0006	4.10	1.4110	5.48	1.7011	6.86	1.9257	8.24	2.1090
1.36	.3075	2.74	1.0080	4.12	1.4159	5.50	1.7047	6.88	1.9286	8.26	2.1114
1.38	.3221	2.76	1.0152	4.14	1.4207	5.52	1.7084	6.90	1.9315	8.28	2.1138
1.40	.3365	2.78	1.0225	4.16	1.4255	5.54	1.7120	6.92	1.9344	8.30	2.1163
1.42	.3507	2.80	1.0296	4.18	1.4303	5.56	1.7156	6.94	1.9373	8.32	2.1187
1.44	.3646	2 82	1.0367	4.20	1.4351	5.58	1.7192	6.96	1.9402	8.34	2.1211
1.46	.3784	2.84	1.0438	4.22	1.4398	5.60	1.7228	6.98	1.9430	8.36	2.1235
1.48	.3920	2.86	1.0508	4.24	1.4446	5.62	1.7263	7.00	1.9459	8.38	2.1258
1.50	.4055	2.88	1.0578	4.26	1.4493	5.64	1.7299	7.02	1.9488	8.40	2.1282
1.52	.4187	2.90	1.0647	4.28	1.4540	5.66	1.7334	7.04	1.9516	8.42	2.1306
1.54	.4318	2.92	1.0716	4.30	1.4586	5.68	1.7370	7.06	1.9544	8.44	2.1330
1.56	.4447	2.94	1.0784	4.32	1.4633	5.70	1.7405	7.08	1.9573	8.46	2.1353
1.58	.4574	2.96	1.0852	4.34	1.4679	5.72	1.7440	7.10	1.9601	8.48	2.1377
1.60	.4700	2.98	1.0919	4.36	1.4725	5.74	1.7475	7.12	1.9629	8.50	2.1401
1.62	.4824	3.00	1.0986	4.38	1.4770	5.76	1.7509	7.14	1.9657	8.54	2.1448
1.64	.4947	3.02	1.1053	4.40	1.4816	5.78	1.7544	7.16	1.9685	8.58	2.1494
1.66	.5068	3.04	1.1119	4.42	1.4861	5.80	1.7579	7.18	1.9713	8.62	2.1541
1.68	.5188	3.06	1.1184	4.44	1.4907	5.82	1.7613	7.20	1.9741	8.66	2.1587
1.70	.5306	3.08	1.1249	4.46	1.4951	5.84	1.7647	7.22	1.9769	8.70	2.1633
1.72	.5423	3.10	1.1314	4.48	1.4996	5.86	1.7681	7.24	1.9796	8.74	2.1679
1.74	.5539	3.12	1.1378	4.50	1.5041	5.88	1.7716	7.26	1.9824	8.78	2.1725
1.76	.5653	3.14	1.1442	4.52	1.5085	5.90	1.7750	7.28	1.9851	8.82	2.1770
1.78	.5766	3.16	1.1506	4.54	1.5129	5.92	1.7783	7.30	1.9879	8.86	2.1815
1.80	.5878	3.18	1.1569	4 56	1.5173	5.94	1.7817	7.32	1.9906	8.90	2.1861
1.82	.5988	3.20	1 1632	4.58	1.5217	5.96	1.7851	7.34	1.9933	8.94	2.1905
1.84	.6098	3.22	1.1694	4.60	1.5261	5.98	1.7884	7.36	1.9961	8.98	2.1950
1.86	.6206	3.24	1.1756	4.62	1.5304	6.00	1.7918	7.38	1.9988	9.00	2.1972
1.88	.6313	3.26	1.1817	4.64	1.5347	6.02	1.7951	7.40	2.0015	9.04	2.2017
1.90	.6419	3.28	1.1878	4.66	1.5390	6.04	1.7984	7.42	2.0041	9.08	2.2061
1.92	.6523	3.30	1.1939	4.68	1.5433	6.06	1.8017	7.44	2.0069	9.12	2.2105
1 94	.6627	3.32	1.1999	4.70	1.5476	6.08	1.8050	7.46	2.0096	9.16	2.2148
1.96	.6729	3.34	1.2060	4.72	1.5518	6.10	1.8083	7.48	2.0122	9.20	2.2192
1.98	.6831	3.36	1.2119	4.74	1.5560	6.12	1.8116	7.50	2.0149	9.30	2.2230
2.00	.6931	3.38	1.2179	4.76	1.5602	6.14	1.8148	7.52	2.0176	9.50	2.2513
2.02	.7031	3.40	1.2238	4.78	1.5644	6.16	1.8181	7.54	2.0202	9.70	2.2721
2 04	.7129	3.42	1.2296	4.80	1.5686	6.18	1.8213	7.56	2.0229	9.90	2.2925
2.06	.7227	3.44	1.2355	4.82	1.5728	6.20	1.8245	7.58	2.0255	10.00	2.3026
2.08	.7324	3.46	1.2413	4.84	1.5769	6.22	1.8278	7.60	2.0281	10.25	2.3279
2.10	.7419	3.48	1.2470	4.86	1.5810	6.24	1.8310	7.62	2.0308	10.50	2.3513
2.12	.7514	3.50	1.2528	4.88	1.5851	6.26	1.8342	7.64	2.0334	10.75	2.3749
2.14	.7608	3.52	1.2585	4.90	1.5892	6.28	1.8374	7.66	2.0360	11.00	2.3979
2.16	.7701	3.54	1.2641	4.92	1.5933	6.30	1.8405	7.68	2.0386	11.25	2.4201
2.18	.7793	3.56	1.2698	4.94	1.5974	6.32	1.8437	7.70	2.0412	11.50	2.4430
2.20	.7885	3.58	1.2754	4.96	1.6014	6.34	1.8469	7.72	2.0438	11.75	2.4636
2.22	.7975	3.60	1.2809	4.98	1.6054	6.36	1.8500	7.74	2.0464	12.00	2.4849
2.24	.8065	3.62	1.2865	5.00	1.6094	6.38	1.8532	7.76	2.0490	12.50	2.5262
2.26	.8154	3.64	1.2920	5.04	1.6134	6.40	1.8563	7.78	2.0516	13.00	2.5649
2.28	.8242	3.66	1.2975	5.04	1.6174	6.42	1.8594	7.80	2.0541	14.00	2.6391

Information Required for Surface and Jet Condenser Installation

To select the proper jet condenser or surface condenser with the auxiliary pumps, it is necessary to know the conditions under which the condenser will operate.

The most necessary information is: The amount of steam to be condensed; the vacuum it is desired to maintain; the temperature of the cooling water.

As in many cases the purchaser cannot state the first conditions definitely, it is necessary to know the other details, which will serve to give a good idea of the requirements of the plant.

In stating the vacuum required, remember that the higher the vacuum, the larger and more expensive the condenser and pumps must be. With any condenser, the smaller the quantity of steam handled, the higher will be the vacuum obtainable, other conditions being equal. Also, the cooler the condensing water or the larger the quantity, the better will be the vacuum obtainable.

Please answer the following questions as fully as possible, and we will give you the benefit of our experience in selecting the proper size condenser.

1. What is the number of pounds of steam to be condensed per hour?
   (a) At ordinary load.
   (b) At peak load.
2. What vacuum is to be maintained?
   (a) At ordinary load.
   (b) At peak load.
3. What is the temperature of the cooling water?
   (a) Under ordinary conditions.
   (b) Under extreme summer conditions.
4. Is there always an ample supply of cooling water available.
5. Is the cooling water fresh, salt, clear or muddy?
6. What is the source of the cooling water?
7. How far from proposed condenser is the source of supply of cooling water located?

8. State size of suction line, if one is already installed.

9. State vertical height cooling water must be lifted by suction.

10. How high above pump must cooling water be delivered, after passing through condenser?

11. If the unit to operate condensing is a steam engine, give the following information:
    (a) Horse power at peak load.
    (b) Type of engine (simple or compound).
    (c) Diameter of cylinder (or cylinders if compound).
    (d) Length of stroke.
    (e) Revolutions per minute.
    (f) Steam pressure at throttle.
    (g) Point of cut off.
    (h) Is engine operating at peak load under conditions specified?
    (i) Do you expect to increase this load after condenser is installed?

12. If the unit to operate condensing is a steam turbine, give the following information:
    (a) Kilowatt rating at peak load.
    (b) Give steam consumption per kilowatt hour at peak load, if possible.

Electrical Data

SECTION FOUR

Electrical Units

Current (I). The strength of current is the rate at which the electricity will flow through a conductor, and is analogous to the rate of flow of water through a pipe in gallons per second. The unit strength of current is called the ampere.

Quantity of electricity (Q). The quantity of electricity that passes through a circuit is comparable to the quantity of water that flows through a pipe, and equals the product of the rate of flow, and the time, that is

$$Q = IT.$$

If I is one ampere, and T is one second, Q is one coulomb, which is the unit quantity of electricity. If 10 amperes flow through a wire, then in 30 seconds $10 \times 30 = 300$ coulombs of electricity will pass.

Electromotive Force (E. M. F.). Electromotive force, or electrical pressure is that which causes electricity to flow in a closed circuit. The unit of E. M. F., which is the volt, is the electrical pressure which will cause a current of one ampere to flow through a resistance of one ohm.

1 Kilovolt = 1000 volts.

1 Millivolt = .001 volts.

Resistance (R). All substances offer a resistance to the passage of electricity through them, and the amount of resistance depends on the substance, and its shape. The resistance of all metals increases with an increase in temperature, while the resistance of carbon and insulating materials, and electrolytic solutions decreases with an increase in their temperature.

The unit of resistance is the ohm. A conductor has a resistance of one ohm, when the pressure required to send a current of one ampere through it is one volt.

Ohm's Law: The relation between current (amperes), pressure (volts), and resistance (ohms), is stated by the famous Ohm's Law. This law is the begining of our scientific knowledge of electricity.

The law is stated as follows: The electric current along a conductor equals the pressure divided by the resistance.

$$\text{Current} = \frac{\text{Pressure}}{\text{Resistance.}}$$

In electric units:

$$\text{Amperes} = \frac{\text{Volts}}{\text{.Ohms.}}$$

$$\text{Volts} = \text{Amperes} \times \text{Ohms}.$$

$$\text{Ohms} = \frac{\text{Volts}}{\text{Amperes}}.$$

Power, Watt, Kilowatt: The flow of an electric current has been likened to the flow of water through a pipe. A current of water is measured by the number of gallons or pounds flowing per minute; a current of electricity by the number of amperes, or coulombs per second. The power required to keep a current of water flowing is the product of the current in pounds per minute by head, or pressure, in feet.

In the same way, the power required to keep a current of electricity flowing, is the product of the current in amperes by the resistance in volts. This gives the power in Watts. One Watt is produced when a current of one ampere flows under a pressure of one volt.

$$\text{Volts} \times \text{Amperes} = \text{Watts}.$$

$$\frac{\text{Volts} \times \text{Amperes}.}{1000} = \text{Kilowatts}.$$

$$\frac{\text{Volts} \times \text{Amperes}}{746} = \text{Horse Power}.$$

1 Myriwatt = 10 Kilowatts.

Work. Commercial Units: In order to compute the amount of work done by a given engine, it is necessary to know the time it has been running, and the power it has been supplying, that is its rate of doing the work. If the power is measured in Horse Power, and the time in Hours, the work done is measured in Horse-Power-Hours, and is the product of the Horse Power by the Hours.

Similarly, if the power is measured in Kilowatts, and the time in hours, the work done is measured in Kilowatt-Hours, and is the product of the Kilowatts by the Hours.

The Horse-Power-Hour, and the Kilowatt-Hour are the commercial units of work.

1 H. P. hour = .746 K. W. hours.
1 K. W. hour = 1.34 H. P. hours.

Two other units of work are also used in computing problems: The mechanical unit is the Foot-Pound. The electrical unit is the Watt-Second, also called the Joule.

1 H. P. hour = 1,980,000 ft. lbs.

1 K. W. hour = 2,654,200 ft. lbs.

1 Joule = .74 ft. lbs.

The following are the electrical units of Work and Power in general use:

Work Units: Watt-second = joule = volt – coulomb.
K. W. hour = 3,600,000 Watt-seconds.

Power Units: Watt = joule per second = volt–ampere = volt–coulomb per second.

Kilowat = 1000 Watts.

$$\text{Kilovolt–Ampere (A. C. unit)} = \frac{\text{Kilowatts}}{\text{Power factor.}}$$

Power Factor

In an alternating current circuit, it is customary to refer to the product of the effective volts and the effective amperes by the name of Apparent Power, and to measure it in volt-amperes. The term "cos ϕ" is then called the Power Factor. Thus when we wish to compute the true or effective power, we find the apparent power (volts x amperes), and multiply it by the Power Factor (cos ϕ). When the voltage and current are in phase, the term (cos ϕ) is unity (1) and the circuit is said to have unity Power Factor.

The Power Factor, then, really indicates what fraction the true or effective power is of the apparent watts, or volt-amperes. It is generally defined by the equation

$$\text{Power Factor} = \frac{\text{effective power}}{\text{apparent power}} = \frac{\text{watts}}{\text{volts x amperes.}}$$

Since the wattmeter always reads the effective power, we have only to attach a wattmeter, an ammeter, and a voltmeter to a circuit to find the Power Factor, and from it the phase difference between the current and the voltage. The product of the volts and the amperes gives the apparent power by which the wattmeter reading is divided to give the Power Factor.

Electrical Equivalents

	A RATE of doing work.	
One Watt	1.	ampere at one volt.
	.7373	foot-pounds per second
	44.238	foot-pounds per minute
	2654.28	foot-pounds per hour.
	.5027	mile pounds per hour.
	.00134	Horse-Power.
	$\frac{1}{746}$	Horse-Power.

	A RATE of doing work.	
One Kilowatt	737.3	foot-pounds per second.
	44238.	foot-pounds per minute.
	502.7	mile-pounds per hour.
	1.34	Horse-Power.

	A RATE of doing work.	
One Horse-Power	550.	foot-pounds per second
	33000.	foot-pounds per minute.
	375.	mile-pounds per hour.
	746.	watts.
	.746	kilowatt.

	A QUANTITY of work.	
One Watt-Hour	2654.28	foot-pounds.
	.503	mile-pounds.
	1.	ampere hour×one vo
	.00134	Horse-Power-Hour.

	A QUANTITY of work.	
One Horse-Power Hour	1,980,000.	foot-pounds.
	375.	mile-pounds.
	746.	watt-hour.
	.746	kilowatt-hour.

	A QUANTITY of current.
One Ampere Hour	One ampere flowing for one hour irrespective of the voltage.
	Watt-hour ÷ volts.

Torque	Force moving in a circle.
	A force of one pound at a radius of one foot.

Horse Power of Motors

To find the actual or brake horse power of an alternating current motor:

$$\text{B. H. P.} = \frac{\text{Volts} \times \text{Amperes} \times \text{Cos } \phi \times \sqrt{N} \times M}{746} \quad (28)$$

M = Motor efficiency.

N = Number of phases.

Cos ϕ = The power factor of the motor.

To find the actual or brake horse power of a direct current motor:

$$\text{B. H. P.} = \frac{\text{Volts} \times \text{Amperes} \times \text{Efficiency of motor.}}{746} \quad (27)$$

Alternating Current

Alternating current is a current consisting of equal half waves in successively opposite directions; it flows back and forth in a circuit with as great a regularity as a piston moves to and fro in a cylinder of a steam engine, but with a much greater rapidity.

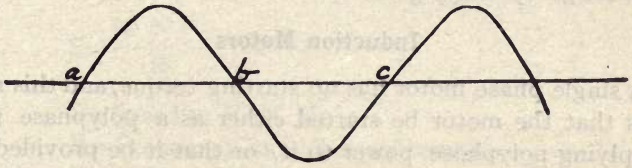


Fig. 94.

Alternating currents are represented by curved lines, as in figure 94, that indicate successive positive values of the current by loops, or half waves a-b above the horizontal line, and the negative values by loop b-c below the horizontal line. Dis-

tances along the horizontal line represent time, the points a-b-c represent instants when the current is zero, from *a* the current increases in a positive direction to a maximum value, falls to zero at *b*, increases to a maximum negative value, and again decreases to zero at *c*, thus completing a cycle. These cycles are continually repeated, usually twenty-five to sixty or more times per second, the number of cycles per second is the frequency of the current, and the time required for the current to complete a cycle is a period. An alternation is a half cycle, and is represented by the curves between a-b, or between b-c. The frequency is generally expressed as the number of alternations per minute, which equals 2x60 x number of cycles per second.

Direct Current

A direct current is a current that flows in one direction; that is, the current never reverses though it may change in value, or pulsate.

Alternating Current Motors

Synchronous Motors

Synchronous Motors are so called because they run in synchronism with the alternator supplying the energy to drive them. Such motors find considerable application in power-transmission systems. They possess the disadvantage that they are not inherently self-starting, so that they have not been very generally adapted for ordinary power requirements. Their speed is constant, regardless of load, which characteristic is often valuable. A characteristic which makes them more particularly adapted to use in power-transmission systems is that their power factor may be controlled and varied through a wide range by varying the field excitation.

Induction Motors

A single phase motor has no starting torque, and this necessitates that the motor be started either as a polyphase motor by applying polyphase power to it, or that it be provided with auxiliary windings to start it as a repulsion motor. All of these methods are used in practice, but the single-phase motor is generally used only in small sizes.

The majority of motors used in the alternating current, are the polyphase type, either 2 or 3 phase.

Induction motors are classified as squirrel cage, or as wound rotor motors according to the type of rotor.

In squirrel cage motors, the inductors are copper bars embedded in longitudinal slots in the laminated steel rotor core, and are connected in parallel to short-circuiting copper rings, one at each end of the rotor.

In the wound rotor, the winding is polar, and the terminals of the windings are connected to a resistance carried on the rotor spider, or through slip rings to an external resistance, which resistance may be cut out when the proper speed is attained. Wound rotor motors are also called slip ring motors. Polyphase squirrel cage motors are used for constant speed service where starting and reversing are infrequent. Their starting torque is relatively small, and a large starting current (2 to 6 times the full load current) is drawn from the line, if the motor must start full load torque.

Squirrel cage motors are adapted to driving centrifugal pumps, generator sets, blowers and any other machinery where a small starting torque is required.

Slip ring or wound rotor motors give about 1½ times the full load torque with approximately 1½ times full-load current, making them suitable for use where a minimum starting current is desirable.

Slip ring or wound rotor motors are adapted to that class of service which requires a heavy starting torque, such as driving power reciprocating pumps, air compressors or other machinery which have to start against a full load.

The synchronous or no load speed of any induction motor can be computed from the equation.

$$S = \frac{60 \times F}{P} \qquad (46)$$

S = Speed in R. P. M.
F = Frequency in cycles per second.
P = Pairs of poles.

Example

What is the speed of a 4 pole 60 cycle motor?

$$S = \frac{60 \times 60}{2} = 1800 \text{ R. P. M.}$$

The actual or full load speed is less than the synchronous or no load speed owing to the losses in the rotor. The difference between the full load and the no load speed is called the slip, see tables on pages 236-237.

Speed of Rotary Field For Different Numbers of Poles and For Various Frequencies

Number of Poles	Speed of Revolving Magnetism, in Revolutions per minute, when Frequency is:											
	25	30	33⅓	40	50	60	66⅔	80	100	120	125	133⅓
2	1500	1870	2000	2400	3000	3600	4000	4800	6000	7200	7500	8000
4	750	900	1000	1200	1500	1800	2000	2400	3000	3600	3750	4000
6	500	600	667	800	1000	1200	1333	1600	2000	2400	2500	2667
8	375	450	500	600	750	900	1000	1200	1500	1800	1875	2000
10	302	360	400	480	600	720	800	960	1200	1440	1500	1600
12	250	300	333	400	500	600	667	800	1000	1200	1250	1333
14	214	257	286	343	428	514	571	686	857	1029	1071	1143
16	188	225	250	300	375	450	500	600	750	900	938	1000
18	167	200	222	267	333	400	444	533	667	800	833	889
20	150	180	200	240	300	360	400	480	600	720	750	800
22	136	164	182	217	273	327	364	436	545	655	682	720
24	125	150	167	200	250	300	333	400	500	600	625	667

Slip of Induction Motors

Capacity of Motor, H. P.	Slip at full load per cent		Capacity of Motor, H. P.	Slip at full load per cent	
	Usual limits	Average		Usual limits	Av'age
⅛	20–40	30	15	5–11	8
¼	10–30	20	20	4–10	7
½	10–20	15	30	3–9	6
1	8–20	14	50	2–8	5
2	8–18	13	75	1–7	4
3	8–16	12	100	1–6	3.5
5	7–15	11	150	1–5	3
7½	6–14	10	200	1–4	2.5
10	6–12	9	300	1–3	2

Full Load Speed of Induction Motors

Number of Poles	Full load speeds of alternating current motors based on 4% slip								
	Cycles								
	25	27	30	33⅓	40	42	50	60	100
2	1440	1560	1730	1920	2300	2420	2880	....	....
4	720	780	865	960	1150	1210	1440	1725	2880
6	480	520	575	640	770	807	960	1150	1920
8	360	390	433	480	575	605	720	862	1440
10	290	310	345	385	460	485	575	690	1150
12	240	260	288	320	385	403	480	575	960
14	205	222	247	275	330	346	412	492	822
16	180	195	216	240	287	302	360	431	720
18	160	171	192	214	256	268	320	384	640
20	142	156	173	192	230	242	287	345	575

Induction Motors
220-440-2200 Volts

Horse Power of Motor	Number of Poles	Synchronous Speed		Approximate Full Load Speed		Efficiency Per Cent				Power Factor Per Cent			
		25 Cycles	60 Cycles	25 Cycles	60 Cycles	½ Load	¾ Load	Full Load	1¼ Load	½ Load	¾ Load	Full Load	1¼ Load
1	4	750	1800	690	1700	74	77	77	76	60	72	80	83
2	6	500	1200	460	1120	82	84	84	86	60	72	78	80
5	6	500	1200	460	1120	84	86	86	85	78	84	86	87
10	6	500	1200	465	1135	85	86	86	84	80	86	89	90
20	6	500	1200	470	1135	87	88	87½	87	82	89	91	90
50	8	375	900	360	850	87	88	88	88	78	86	89	90
100	10	300	720	288	690	89	90	90	90	83	89	91	91
200	12	....	600	....	575	91	92½	92	91	85	91	92	91

Note: 2200 volt motors are seldom made in sizes under 20 Horse Power.

Ampere Ratings of A. C. Motors

H.P.	Single Phase				Two Phase					Three Phase				
	Volts				Volts					Volts				
	110	220	440	550	110	220	440	550	2200	110	220	440	550	2200
⅛	....	....	....	....	....	....	....	....	....	....	....	....	....	....
¼	4.0	2.0	1.0	.8	....	....	....	....	....	....	....	....	....	....
½	7.5	3.75	1.9	1.5	3.2	1.6	.8	.6	....	3.6	1.8	.9	.7	....
¾	10.	5.	2.5	2.	....	....	....	....	....	....	....	....	....	....
1	12.5	6.25	3.2	2.5	6.2	3.0	1.6	1.2	....	7.2	3.6	1.8	1.4	....
1½	18	9	4.5	3.6	....	....	....	....	....	....	....	....	....	....
2	24	12	6	4.8	10.	5.0	2.5	2.0	....	11.6	5.8	2.9	2.3	....
3	34	17	8.5	6.8	14.2	7.0	3.6	2.8	....	16.4	8.2	4.1	3.3	....
4	43	22	11	8.6	19	9.0	5	4	....	22	11	6	4.5	....
5	55	28	14	11	24	12	6	5	....	28	14	7	6	....
7½	80	40	20	16	33	17	9	7	....	39	19	10	8	....
10	105	53	25	21	47	23	12	10	....	54	27	14	11	....
15	145	73	36	29	70	35	18	14	....	81	40	20	16	....
20	200	100	50	40	90	45	23	18	....	104	52	26	21	....
25	250	125	63	50	112	56	28	23	6	130	65	33	26	7
30	295	148	74	59	132	66	33	26	7	152	76	38	30	8
40	370	185	93	74	175	88	44	35	9	202	101	51	41	10
50	....	....	....	....	216	108	54	43	11	250	125	63	50	13
60	....	....	....	....	260	130	65	52	13	300	150	75	60	15
75	....	....	....	....	324	162	81	65	16	375	187	94	75	19
100	....	....	....	....	432	216	108	86	22	500	250	125	100	25
125	....	....	....	....	540	270	135	108	27	625	312	156	125	31
150	....	....	....	....	648	324	162	129	32	750	375	188	150	37
175	....	....	....	....	758	379	189	152	38	875	438	219	175	44
200	....	....	....	....	864	432	216	173	43	1000	500	250	200	49
250	....	....	....	....	1080	540	270	216	54	1250	625	313	250	63
300	....	....	....	....	1298	649	325	259	65	1500	750	375	300	75
400	....	....	....	....	1730	865	433	346	87	2000	1000	500	400	100
500	....	....	....	....	2160	1080	540	432	108	2500	1250	625	500	125

Direct Current Motors

Direct current motors are classified according to their winding into series-wound, shunt-wound and compound-wound.

The series-wound motor is adaptable to variable speed work, such as elevator service, crane service, etc., on account of its speed-torque characteristics. The series-wound motor develops almost any torque demanded of it, but with a reduction in speed, so that it is especially adapted to hard work where constant speed is not essential.

The shunt-wound motor is used almost universally for constant speed service, for it maintains approximately constant speed regardless of load. The shunt-wound motor has a rather low starting torque, and is adapted to such uses as driving blowers, and other machinery in which the starting torque required is small.

The compound-wound motor which is a combination of the series-wound and shunt-wound motors has an advantage of a large starting torque, and is particularly adapted to driving power reciprocating pumps, centrifugal pumps, air compressors, and other machinery.

Ampere Ratings of D. C. Motors

H. P.	Rating of D. C. Motors — Full Load Amperes			Ampere Capacity of Fuses for Motors		
	115 Volts	230 Volts	500 Volts	115 Volts	230 Volts	500 Volts
1/8	1.4	.70	.35	5	5	5
1/6	1.8	.90	.46	5	5	5
1/4	2.2	1.1	.54	5	5	5
1/2	4.3	2.2	1.0	7	5	5
3/4	6.2	3.1	1.5	10	5	5
1	8.0	4.0	1.9	15	7	5
2	16	8.0	3.8	25	15	7
3	24	12	5.5	30	20	10
4	32	16	7.2	50	25	10
5	38	19	9.0	50	30	15
7 1/2	58	29	13	80	50	20
10	75	38	17.5	100	50	25
12 1/2	94	47	22	125	70	30
15	112	56	26	150	70	35
20	148	74	34	200	100	50
25	185	93	43	275	125	60
30	220	110	51	275	150	70
35	255	125	60	325	175	80
40	285	145	67	400	200	90
45	320	160	75	400	200	100
50	350	175	83	450	225	125
60	420	210	99	550	275	125
75	520	260	122	800	325	175
90	625	315	145	.....	.....	.....
100	700	350	160	.....	.....	.....
125	875	440	200	.....	.....	.....
150	1050	525	240	.....	.....	.....
175	1225	615	280	.....	.....	.....
200	1400	700	320	.....	.....	.....

Selection of Motors and Controllers

The selection of the proper type of motor and controlling equipment to be used with centrifugal pumps, air compressors, power pressure pumps and power vacuum pumps is very essential.

While the local conditions may govern to some extent the type of motor and controlling equipment to use, the following paragraphs give the customary types of motors and controlling equipment to use with machinery, such as we manufacture.

For direct current motors, we recommend the compound-wound type for driving centrifugal pumps, air compressors, power pressure pumps and power vacuum pumps.

For alternating current motors, we recommend the squirrel cage type for driving centrifugal pumps, and the wound-rotor or slip-ring type motors for driving power pressure pumps, power vacuum pumps and air compressors.

For reciprocating machinery such as power pressure pumps and power vacuum pumps, which are to operate under automatic control, we recommend that the motor be a size larger than is regularly required. This is due to the fact that under automatic control, the machinery has to start up under full load, which requires a very heavy starting torque.

On the following pages is illustrated and described the proper electrical control equipment to be used in connection with centrifugal pumps, power pressure pumps, air compressors and power vacuum pumps. The electrical control equipment as given, covers both direct, and alternating current apparatus, and includes the manual starters, as well as the self starters.

Electrical Control Equipment

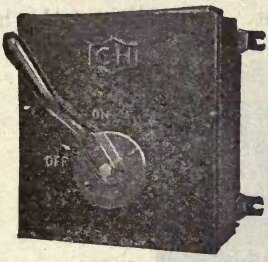
(*The Cutler-Hammer Mfg. Co.*)

For Centrifugal Pumps, Air Compressors, Power Pressure Pumps, and Power Vacuum Pumps

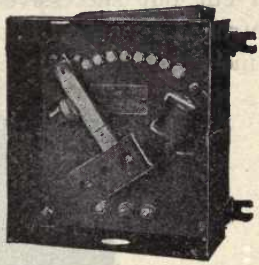
Motors: Direct Current, Compound Wound
Manual Starters

Standard duty motor starters with low voltage protection, Bulletin 2111 up to, and including, 35 HP–115 volts, and 50 HP–230 or 550 volts.

The standard duty starter consists of a sliding contact type starting panel, with armature resistor self-contained, mounted on a wall type frame. The low voltage protection feature stops the motor when the voltage drops or fails altogether and prevents it from restarting when normal voltage is restored. This protects both the machine and operator against the motor's starting unexpectedly.



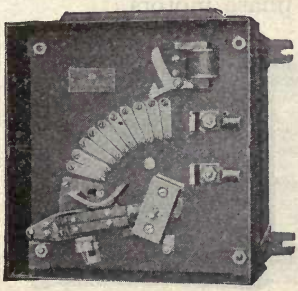
Bulletin 2111.



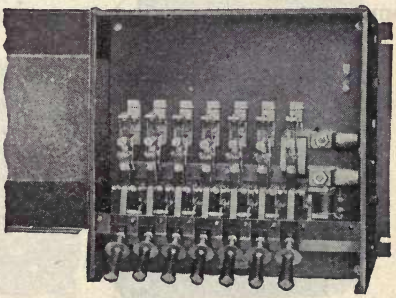
With cover removed, to show construction.

Heavy duty manual starters with low voltage protection, Bulletin 2131. Sliding contact type, similar to Bulletin 2111, except greater capacity resistor, up to and including 20HP–115 to 550 volts. Larger sizes, up to and including 125HP, 115 volts and 200HP–230 or 550 volts are of the multiple switch type.

The multiple switch starter consists of a slate front containing a number of levers which, when closed in sequence, function similarly to the sliding contact type starter. The levers are so interlocked, that they can only be closed in sequence. All starters up to 75HP,–115 volts and 100HP,–230 and 550 volts are in wall type enclosing cases. Larger sizes are arranged for floor mounting.



Bulletin 2131, with cover removed.

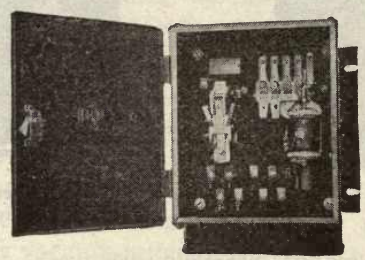


Bulletin 2131 Multiple Switch Type, with cover removed.

D. C. Automatic Motor Starters

Time limit type automatic starters, Bulletin 6106, up to 60HP–115 volts, 125HP–230 volts and 200HP–550 volts. They can be controlled by pushbutton master stations, snap switch, float switch, pressure regulator, and other accessories.

The accelerating movement consists of a number of fingers which cut out successive steps of resistor. This movement is controlled by an oil filled dash pot. In addition, the larger sizes include a magnetic main line contactor.



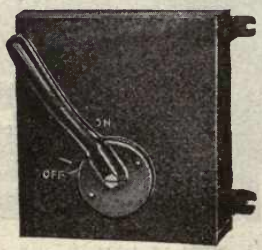
Bulletin 6106.

Alternating Current

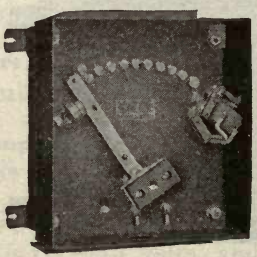
Motors: Single Phase, Repulsion Type
Manual Starters

Single phase motor starters, Bulletin 9111, with low-voltage protection, up to and including 20HP–110 volts and 30HP–220 volts.

These single phase motor starters are similar in construction to the Bulletin 2111, but are arranged for alternating current service. They consist of a slate panel with the sliding contact type starting mechanism and can be used with either commutator or repulsion type single phase motors.



Bulletin 9111



With cover removed.

Across-the-line type manual starters, Bulletin 9115, with thermal overload cutouts, up to and including 1 HP-110 volts, 3 HP-220 volts and 5HP-440 or 550 volts—single phase.

These starters consist of a 3 pole manually operated contactor with two thermal overload cutouts, mounted on a slate panel in an enclosing case. The operating handle and locking levers are on the outside of the case. Contacts are of new roller type-double contact. For single phase service the center pole is left "dead."

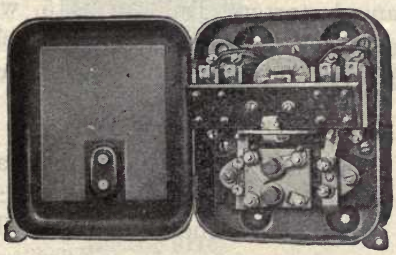
Bulletin 9115.

A. C. Automatic Motor Starters

Across-the-line type self-starters, Bulletin 9586, up to 15HP–110 volts, 30HP–220 volts and 40HP–440 or 550 volts.

These starters consist of a positive acting, three pole magnetic contactor and C-H Thermal Overload Relays, mounted on a panel in a split case. For single phase service, the center pole is left "dead." The controller is operated by a remote control pushbutton master switch or by any type of single pole switch such as a snap switch, pressure regulator, etc.

The C-H Thermal Overload Relays provide ideal protection at all times.

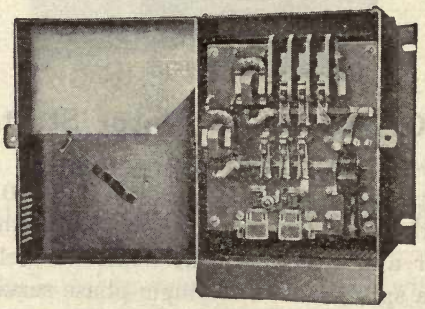


Bulletin 9586 Type AAA

Bulletin 9586 Types AA or A.

Primary resistor type automatic starters, Bulletin 9605, up to and including 15HP–220 volts, and 30HP–440 or 550 volts, single phase.

This is a three pole starter, providing one step of primary resistor on starting. It consists of two, three-pole contactors and C-H Thermal Overload Relays mounted on a slate panel in an enclosing case. For single phase service the center pole is left "dead." The first contactor closes the circuit with resistance inserted and the closing of the second, which is timed by an oil dash-pot, cuts out the resistor, connecting the motor directly to the line. The overload relays protect the motor from overload at all times and when tripped can be reset from the outside of the case.



Bulletin 9605.

Motors: A. C. 2 or 3 Phase Squirrel Cage

Manual Starters

Across-the-line type manual starters, Bulletin 9115, with thermal overload cutouts, up to and including 3 HP–110 volts, 5 HP–220 volts and 7½ HP–440 or 550 volts—two or three phase.

These starters consist of a three or four pole manually operated roller type contactor and two thermal overload cutouts, mounted on a slate panel in an enclosing case. The four pole contactor is for two phase, four wire service only.

Operating and locking levers are on outside of case. Thermal overload cutouts provide overload protection at all times.

Bulletin 9115.

Panel type, fused starting switches, with low-voltage protection, Bulletin 9116, up to and including 3HP–110 volt and 5HP–220 to 550 volts.

The panel type fused starter consists of a three-pole fully enclosed switch, fused and so designed that in starting the fuses are not in circuit, but when in the running position the fuses are in the circuit for protecting the motor. The cover can be lowered, for renewing fuses. This starter also provides low-voltage protection.

Bulletin 9116.

With cover removed to show construction.

Primary resistor type manual starters, Bulletin 9118, for motors up to and including 7½HP–110 volts and 10HP–220 to 550 volts.

This starter is for use on motors that cannot be thrown directly across the line, and which do not require the more costly auto-transformer starter (compensator). It is a panel type starter, similar to Bulletin 9116, except that it provides one step of resistance in each phase of the primary circuit when starting. This cuts down the large current inrush to the motor on starting. It is provided with running fuses, provides low-voltage protection and is fully enclosed, being operated by a lever on the outside of the case.

Bulletin 9118.

Auto-transformer starters, Bulletin 9141, up to and including 25HP-110 volts and 125HP–220 to 2200 volts.

This type of starter in the smaller sizes is self-contained, consisting of a metal case containing two auto-transformers, switching mechanism, overload and low-voltage protection features. The switching mechanism which operates under oil, functions to connect the starting transformer to the power lines, also connecting the motor to taps on the transformers for starting, without drawing excessive current from the line. When the motor is approximately up to speed, operation of the lever serves to disconnect the starting transformers and connects the motor directly to the line.

The low-voltage protection feature can be operated mechanically at the starter or by means of pushbuttons at a remote point when it is desired to stop the motor. The overload release feature will not trip on small momentary overloads.

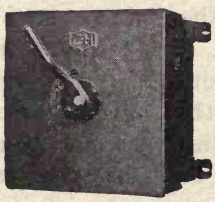
However, it does disconnect the motor if an overload occurs which may be injurious to the motor or machine. It also functions to shut down the motor on partial failure of power lines, that is, failure of one line which would allow the motor to operate on single phase. On the larger sizes the transformers are not mounted in the same case with the switching mechanism, but the operation and functions are the same in all cases.

Bulletin 9141.

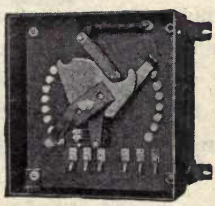
Manual Starters, Slip-Ring Motors

Secondary resistor type, Bulletin 9126, up to and including 25HP–280 volts and 50HP–300 volts rotor limitations.

These starters are of the sliding contact type, arranged to cut out starting resistance in the rotor circuit. They are mechanically released, and have no "dead" point. To stop the motor, it is necessary to return the lever to the starting point and open the main line switch. If a magnetic main line switch is used, it can be interlocked with the starter so that it closes on the first starting point and opens when the lever is returned to the original position. Care should be taken so as not to exceed the rotor limitations given in the Bulletin.



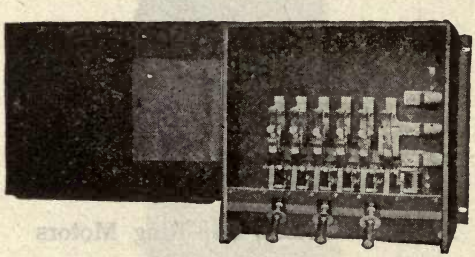
Bulletin 9126.



With cover removed.

Multiple switch motor starter, Bulletin 9131, up to and including 2000 HP.

These starters are of the secondary resistor type and consist of a slate panel containing a number of levers, which, when closed in sequence, cut out steps of resistance in the rotor circuit to bring the motor up to speed. The levers are so interlocked that they can only be closed in sequence. Up to 200HP, the resistor is in the same enclosing case with the panel. The larger sizes have the resistor separately mounted.



Bulletin 9131.

A. C. Automatic Starters, 2 or 3 Phase Squirrel Cage Motors

Across-the-line type automatic starters, Bulletin 9586, for capacities up to 150 amperes, at 550 volts.

These starters consist of a positive acting three or four pole magnetic contactor, and C-H thermal overload relays mounted on a panel in a split case. The smallest size, for motors up to 5HP, has roller type contacts and is provided with a two button control switch in the cover. Larger sizes have finger type contacts and are supplied with one pushbutton master switch for three wire control. C-H Thermal Overload

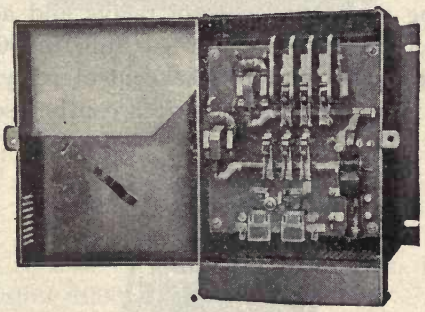
Relays provide ideal protection. When a relay is tripped, it can be reset by pushing a button on the outside of the case.

Bulletin 9586, Type AAA.

Bulletin 9586,
Types AA and A.

Primary resistor type automatic starters, Bulletin 9605, up to and including 30HP–220, 440 or 550 volts, 2 or 3 phase.

These starters are identical with those listed for single phase service, except that for polyphase motors, all three poles are used. They can be controlled by any type of three wire or two wire master switch.

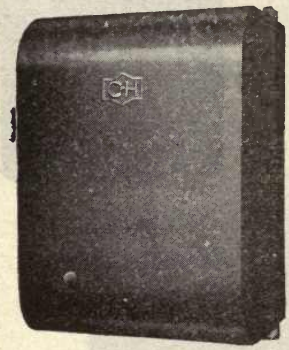


Bulletin 9605.

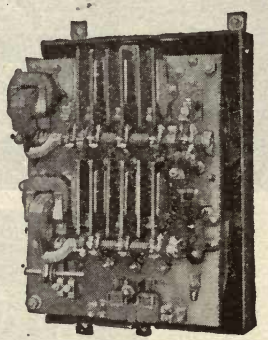
Auto transtormer type automatic starter, Bulletin 9621, up to and including 30HP–220 volts and 40HP–440 or 550 volts.

These starters consist of a three pole and a five pole mag-

netic contactor, and C-H Thermal Overload Relays mounted on a slate panel in a split enclosing case. On starting, the five pole contactor closes, connecting the motor to the line, thru the auto transformer. After a definite time interval, determined by an oil dashpot, the three pole contactor closes, connecting the motor across-the-line. Both low voltage and over-load protection are provided.



Bulletin 9621.



With cover removed.

Transformer type automatic starters, Bulletin 9622, for 2200 volt circuits, up to and including 400HP.

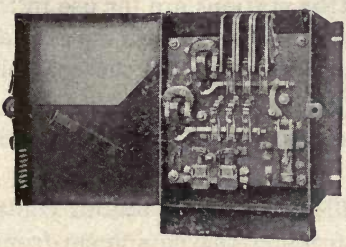
These starters consist of a three pole and a five pole oil immersed magnetic main line contactor, a solenoid operated dash pot timing relay, two control relays and an auto transformer. The operation is similar to Bulletin 9621, the large contactor being operated thru the control relays. Any type of two wire, or three wire master switch can be used for control.

Bulletin 9622.

A. C. Automatic Starters, 2 or 3 Phase Slip-Ring Motors

Secondary resistor type automatic starter, Bulletin 9633, up to 25HP–110 volts, 50HP–220 volts and 75HP–440 or 550 volts. These starters are for wall mounting. They consist of a three pole magnetic main line contactor and one or more three pole accelerating contactors. C-H thermal overload relays provide ideal overload protection at all times. Low-voltage release is also provided.

On starting, the accelerating contactors close in rotation, cutting out steps of resistance in the secondary circuit. The main line contactor then closes, connecting the motor directly to the supply lines. In selecting a starter of this type, care should be taken so that the secondary current rating is not exceeded. They can be controlled by any type of two or three wire master switch.

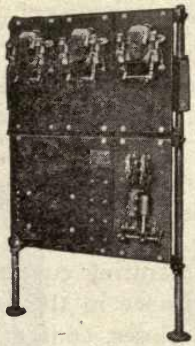


Bulletin 9633.

Secondary resistor type automatic starters, Bulletin 9638, up to and including 1000HP–2200 volts.

These starters operate similar to the Bulletin 9633 starters, except that they are built for higher voltages—up to 2200 volts. The main line contactor is a three pole oil-immersed magnetic contactor and the accelerating contactors are two pole. Control relays govern the closing of these contactors. They can

be controlled by any type of two wire or three wire master switch.



Bulletin 9638.

C. H. Accessories

Pushbuttons

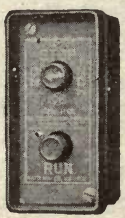
These pushbutton stations are for use in the control circuit of alternating or direct current automatic starters, to control the various operating functions of the controller.

Bulletin 10250H30 is a single button switch, intended for use as an auxiliary in connection with control stations of the two button type.

Bulletin 10250H26 is a two button control station, used for starting and stopping the equipment from a remote point. The "stop" button can be locked in the down position, providing a "safe" feature.

Bulletin 10250H56 is a two button control station used for starting and stopping from a remote point. It does not provide the "safe" feature.

Bulletin 10250H30



Bulletin 10250H26

Bulletin 10250H56

Diaphragm type pressure regulator, Bulletin 10001 for pressures above atmosphere only.

This pressure regulator is of the diaphragm type, single pole, for handling control circuits only, having a maximum capacity of 175 pounds and arranged for use with any of the self-starters.

These regulators are manufactured in four different sizes for various pressures. In selecting these regulators, be careful that the ranges between opening and closing are not exceeded.

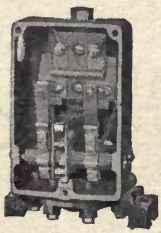
Bulletin 10001.

With cover removed.

Diaphragm type pressure regulators, two pole, Bulletin 10004.

This is a two pole, diaphragm type, regulator, for use with small A.C. or D.C. motors which can be connected directly to the line to start. It handles the main line circuit, and no other starter is required. An unloader device can be provided for air compressor work. This device relieves the back pressure against the compressor during starting. Pressure limitations or HP ratings given in the Bulletin should not be exceeded.

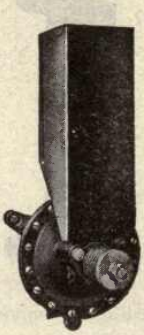
Bulletin 10004.



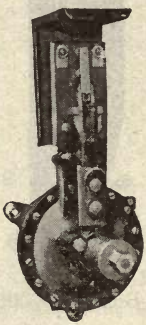
Witn cover removed.

Diaphragm type vacuum regulators, Bulletin 10005.

These regulators are similar in design to the pressure type, Bulletin 10001 but are arranged for pressures below atmosphere only, with a maximum vacuum range of 28 inches of mercury. These vacuum regulators are suitable for use with any of the automatic starters, but as in the case of pressure regulators, it is necessary to see that the maximum and minimum range between opening and closing does not exceed the limits tabulated in the bulletin.



Bulletin 10005.



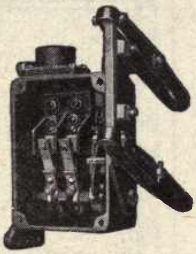
With cover removed.

Gauge type pressure regulators, with relay, Bulletin 10013. These gauge type pressure regulators are for use with systems having a greater pressure range than can be handled by the diaphragm type, and are suitable for use with any of the "A.C" or "D.C." automatic starters.

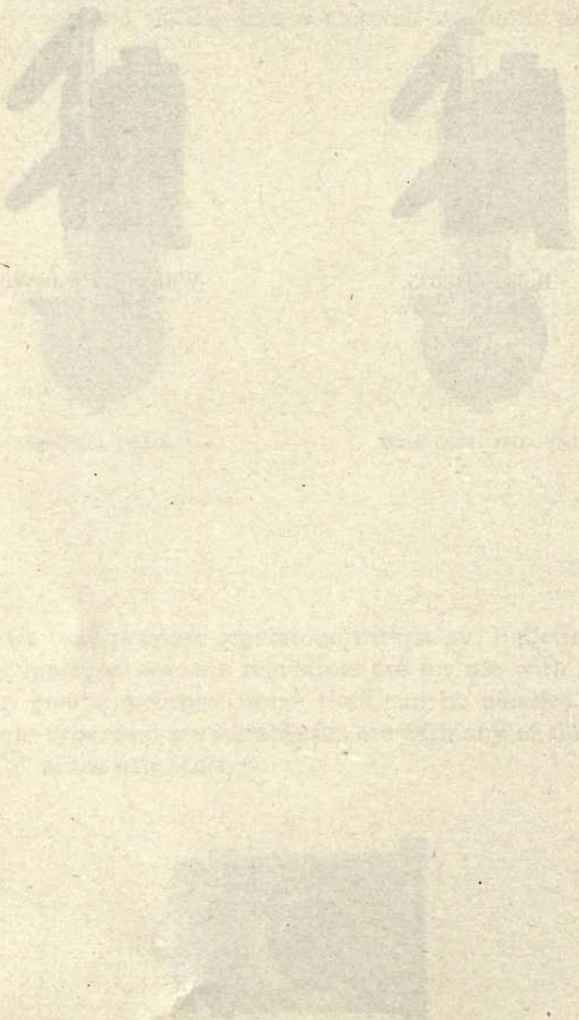
Bulletin 10013.

Enclosed type float switches, Bulletin 10036, two or four pole. These switches can be used as pilot devices in connection with A.C. or D.C. automatic starters, or they can be used to connect small A.C. motors directly across-the-line. Six types of mounting can be supplied, to suit local conditions. These float switches can be arranged for either tank or sump operation.

Bulletin 10033.



With cover removed.



Pumps

Direct Acting Steam Pumps

SECTION FIVE

Direct Acting Steam Pumps

The direct-acting steam pump is one in which the water end is placed centrally, or directly in line with the steam end. The water piston and the steam piston are placed on the same rod, and both operated together independently of any crank movement.

. Direct-acting steam pumps are classified according to the water end as follows:

Single Double Acting: This type, which is illustrated in figure 95, is the simplex design, having one steam piston operating one water piston. The water cylinder is double acting, in other words each stroke of the piston causes a filling at one end of the pump cylinder, and a discharge at the other end of the cylinder.

Duplex Double Acting: This type illustrated in figure 98, consists of two steam pistons operating two water pistons. The steam and water cylinders are operated side by side, and the steam valve of one side is actuated by the companion pump. The water cylinders are double acting, in other words each stroke of the piston causes a filling at one end of the pump cylinder, and a discharge at the other end of the cylinder.

The above pumps may also be sub-divided as follows:

Horizontal Double Acting Pumps

Piston Packed Pumps.

Outside Center-Packed Plunger Pumps.

Outside End-Packed Plunger Pumps.

Vertical Double Acting Pumps

Piston Packed Pumps.

Outside Center-Packed Plunger Pumps.

Outside End-Packed Plunger Pumps.

Direct acting steam pumps are classified according to the steam end into simple and compound.

The above types comprise the direct-acting steam pumps, which are generally encountered. There are other types, but they are special, and not considered of sufficient importance to enumerate.

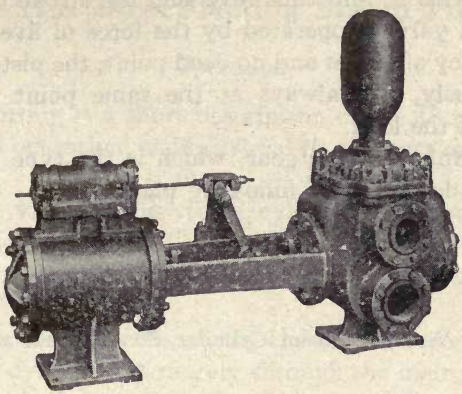


Fig. 95

Single Pumps

The characteristic feature of single pumps is the valve motion, or mechanism introduced to reverse the motion of the pump.

This valve motion generally consists of an auxiliary valve, figure 96, mechanically operated by the steam piston and controlling an auxiliary piston, which in turn operates the main valve. The latter operates the steam piston, thus completing the cycle. These four elements can always be recognized in a single pump.

There are numerous types of valve motions employed on single pumps, but the most satisfactory is that type in which the steam piston is controlled by a slide valve, the valve itself being operated by an auxiliary steam piston working in its own chest, and the auxiliary piston being moved by the direct application of steam pressure.

The admission of the steam to the end of the auxiliary steam piston in the chest is controlled by a flat-faced auxiliary slide valve in the chest, which is actuated by an external valve mechanism from the piston rod. As the pump reaches the end of the stroke, steam is admitted by the auxiliary valve to one end of the auxiliary piston, while the other end is put into communication with the exhaust at the same time, and the difference of pressure in the two ends causes the auxiliary piston to move instantaneously the full distance of its travel, carrying the main

slide valve with it, and thus reversing the stroke of the pump. As the main valve is operated by the force of live steam, there is no hesitancy of action and no dead point, the piston is reversed instantaneously, and always at the same point of its stroke regardless of the load.

The Burnham valve gear, which is the type referred to is described in detail on the following pages.

Burnham Steam Pump

Sectiona. Views of Steam Cylinder, Steam Chest and Valves.

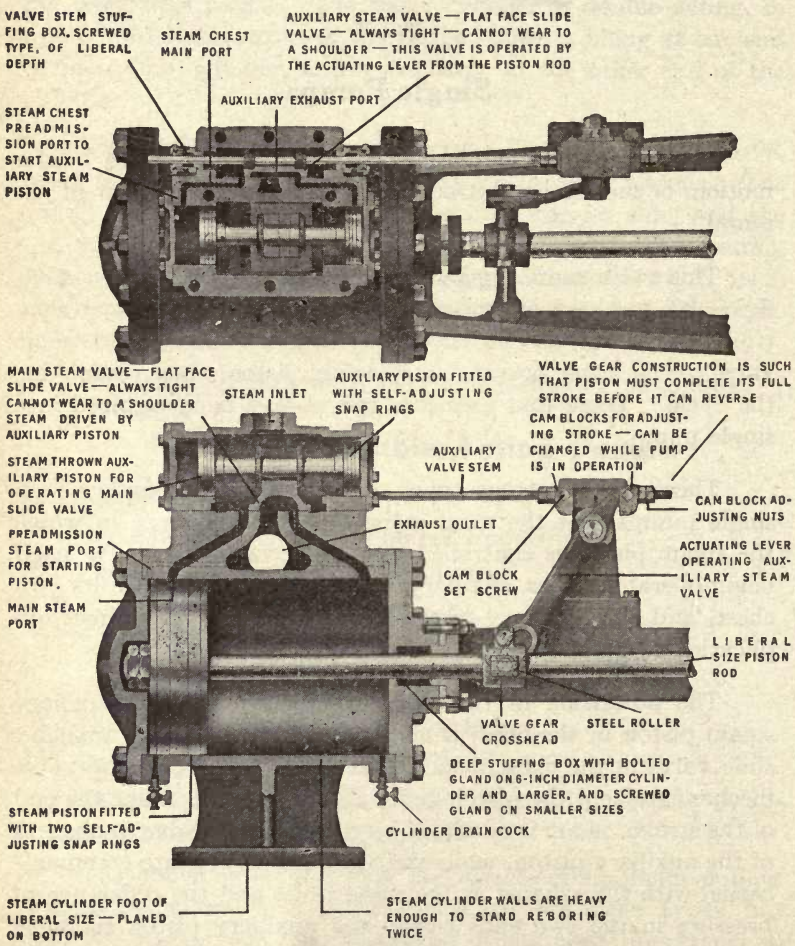


Fig. 96

Burnham Direct-Acting Steam Pumps

Operation

The following is a short description of the Burnham Steam Valve and its operation, as illustrated by the cuts on the preceding page.

The top view of the steam cylinder shows the auxiliary valve and chest in section.

The lower view shows a vertical section through the steam cylinder.

Live steam enters the steam chest at the top and is admitted to the cylinder alternately through the main steam ports.

At the beginning of the stroke the main steam port is covered by the piston as shown in the cut. A preadmission port is provided which admits only enough steam to give the steam piston an easy start, but when the steam piston has moved far enough to uncover the main port, it receives the full steam pressure and moves at its normal speed until it covers the main port at the other end of the cylinder, when it traps the remaining exhaust steam in the cylinder and thus forms a cushion, giving the steam piston an easy stop.

The valve gear is positive in action, and is operated by the actuating lever moved by a roller attached to the piston rod. This lever alternately moves the cam blocks both of which are fastened to the auxiliary valve stem, which in turn moves the auxiliary valve in the direction opposite to the motion of the piston.

When the steam piston completes its stroke, the actuating lever moves the auxiliary valve, opening first the chest preadmission port, then the chest main port, admitting live steam to one end of the auxiliary piston and at the same time opening the auxiliary exhaust at the opposite end, thus causing the auxiliary piston, which carries the main valve, to move, reversing the motion of the pump.

The cam blocks are independently adjustable on the auxiliary valve stem enabling the engineer to make the piston run as close to the heads as he desires, and to make adjustment to compensate for wear.

The advantages of this valve gear are: a momentary pause of the piston at the end of each stroke, causing the water valves to seat quietly without shock or jar; a slow initial movement of the piston, whereby the water columns are started gradually, relieving the pump and piping of undue strains; a

steam pressure on the main steam piston proportioned to the amount of work that it has to do; and immunity from damage in case of accident.

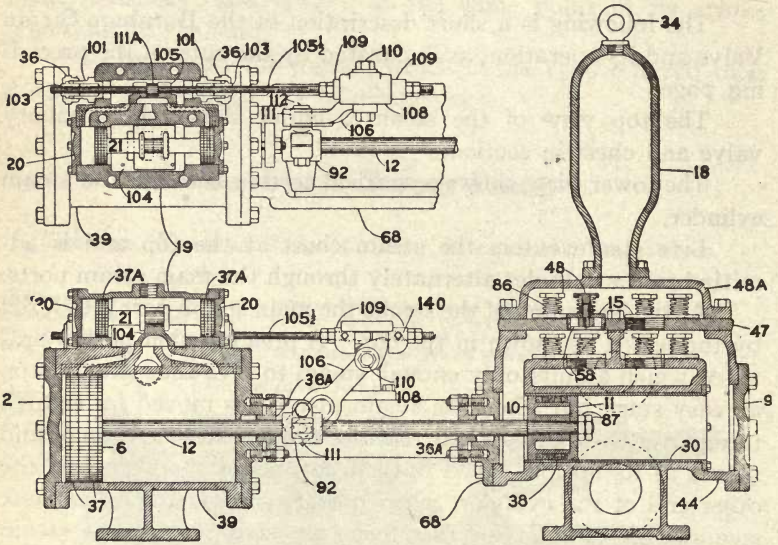


Fig. 97
Sectional View of Burnham Pump.

Directions for Setting the Burnham Valve

All Burnham pumps are carefully tested at the factory under working conditions and the valve gear is properly set. If it becomes necessary to readjust the valve gear, proceed as follows:

On yoke 68 (see figure 97), upon which moves the piston-rod guide 92, will be found a mark at each end, indicating the extreme travel of the piston.

If the pump does not run as close to the mark as practical, loosen the nuts on the valve stem and the set screw in cam block 109 on the opposite side (of the actuating lever 106) from which it is desired to lengthen the stroke, and move the cam block away from the point of contact of actuating lever 106.

This will allow the piston to move farther before opening the valve.

It will be found that by moving this cam block $\frac{1}{16}$ of an inch, it makes quite a perceptible difference in the piston travel, according to the size of pump to be adjusted.

If the pump should travel too close to the marks, which would cause it to hesitate and stop at the end of stroke, then move these cam blocks 109 toward the point of contact of actuating lever 106.

Always move the cam blocks on the opposite side of lever from which it is desired to change the stroke.

In all cases the piston should make as long a stroke as possible and give the required speed to do the work.

To locate the marks on the yoke, indicating the extreme travel of the piston, move the steam piston to one end till it strikes the cylinder head and mark with a prick punch the yoke on which the piston rode guide 92 rides. Repeat this operation on the other end, and use these marks to adjust the valve as described.

Advantages of Single Pumps

The single pump is the most desirable type of direct acting pump because of its simplicity, reliability and economy in operation and maintenance.

The single pump is simple in construction and has a comparatively few number of moving parts and packed joints, with the result that there is a large saving in friction. The few number of moving parts required in the single pumps means less wear and less liability to accidents and slippage and shutdowns for repairs, as well as entailing less care on the part of the operating engineer to see that the parts are in the proper running condition.

It is a well-known fact that the chief sources of loss in any steam-actuated machine are those by direct radiation through the walls of the cylinders, and by condensation of the steam on the walls during admission, with subsequent re-evaporation during exhaust. Such heat losses mean of course wasted energy. As the radiating surface increases, the loss of energy increases with it, and as a single pump has a minimum radiating surface, the heat losses are a minimum.

In the direct-acting steam pump, all steam used to fill the port passages of the cylinder, and the clearances at the end of the stroke, is wasted as it is rejected to the exhaust without having done any work. Clearance is a necessary evil, so it is made as small as possible with due regard to proper running of the pump. Single pumps are made with but one steam port at each end of the cylinder, and this reduces the wasted steam space to a minimum.

The greatest advantage of the single pump as regards steam consumption lies in the fact that with its valve motion, it has to complete its full stroke before it can reverse. This means that the waste steam space at the end of the stroke which is the source of greatest loss in a pump is minimized.

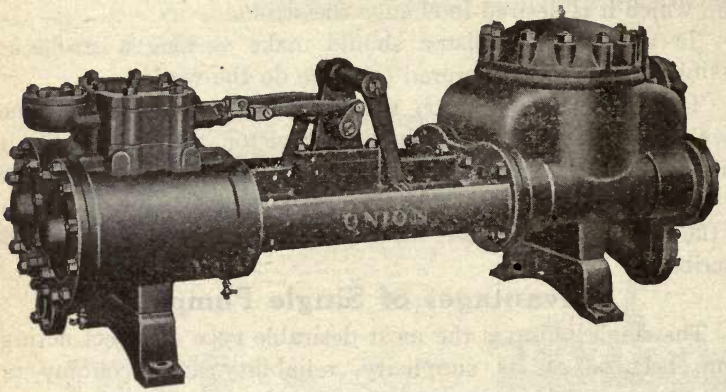


Fig. 98.
Duplex Piston Pump.

Duplex Pumps

Duplex pumps are characterized by the arrangement of cylinders and type of valve gear. The universal arrangement is to place two pumps side by side, the main steam valve of one side being actuated through a system of levers, rods and links from the piston rod of the other side. In designating the side of a duplex pump, it is customary to call the right side when standing at the steam end facting the water end of the pump, the "Right Hand Side," and that to the left, the "Left Hand Side."

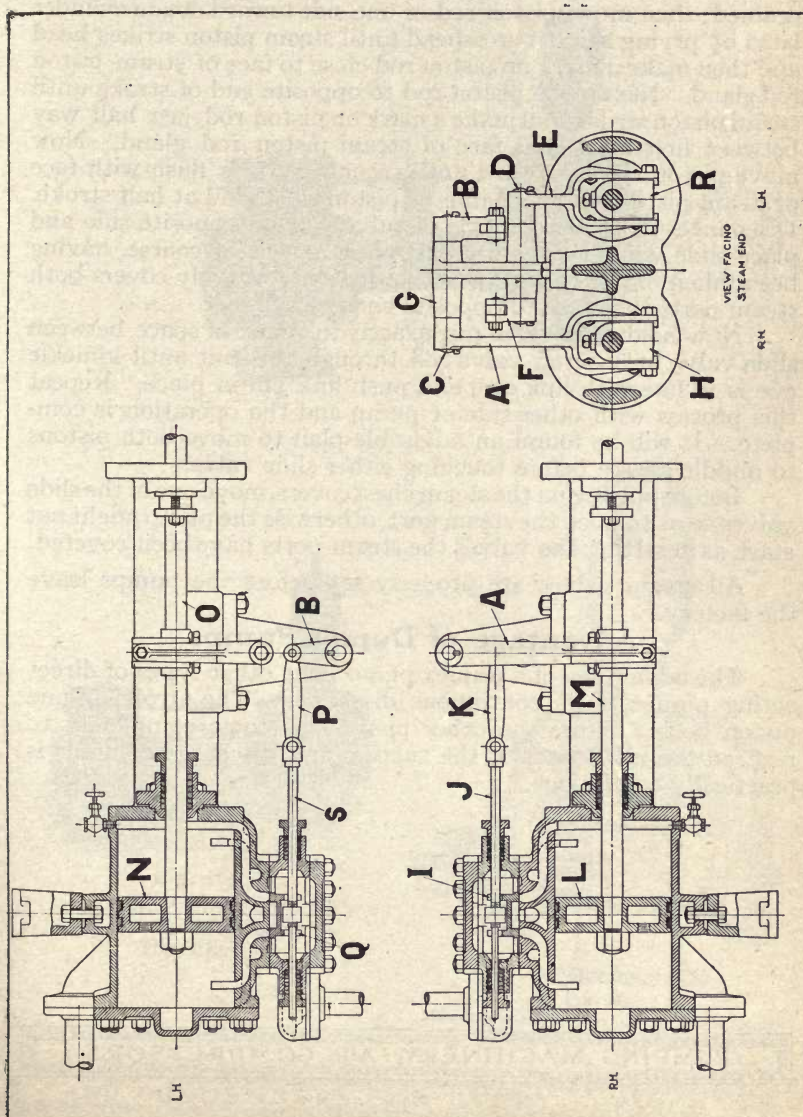
Valve Motion

The sectional view, figure 99, which illustrates the Union duplex pump, shows the type of valve gear generally employed on duplex pumps. The steam valves Q and I are ordinary D slide valves. The valve motion is transmitted from the pistons to the valves by two rock-shafts C and D mounted on the valve gear bracket G. To the right hand end of the upper rock-shaft C is fitted a long arm A, the lower end of which is attached to the crosshead H on the piston rod. To the left hand end of the rock shaft C is fitted the long crank B attached to the valve rod S by means of a coupling and link P.

To the left hand end of the lower rock-shaft D is fitted a short arm E. To the right hand end of the lower rock shaft D is fitted the short crank F attached to the valve rod J by coupling and link K.

Thus the valve I on the right hand pump is operated through the valve rod J, link K, crank F, rock-shaft D, arm E, piston rod O, and piston N. The valve on the left hand pump is operated through the valve rod S, link P, crank B, rock-shaft C, arm A, piston rod M, and piston L.

On the duplex pump, the valve motion is such that one side finishes its stroke, and waits for its valve to be moved by the other side of the pump, before it can start on its return stroke. This pause allows the water valves to seat quietly and obviates shocks. As one or the other of the steam valves is always open, there is no dead center, and the pump will start whenever the steam is turned on.



Directions for Setting Steam Slide Valves of Duplex Pumps

To set the Slide Valves of Duplex Pumps without outside adjustment.—First open drip cocks so that steam cylinders will be drained; then move piston rod of one side toward steam cylinder head by prying against crosshead until steam piston strikes head and then make a mark on piston rod close to face of steam piston rod gland. Next move piston rod to opposite end of stroke until steam piston strikes and make a mark on piston rod just half way between first mark and face of steam piston rod gland. Now move piston rod backward until second mark is flush with face of steam piston rod gland and the piston will stand at half stroke. Disconnect link from knuckle of valve rod on opposite side and place slide valve in steam chest, chest cover, of course, having been taken off for this purpose, so that valve exactly covers both steam ports that lead to opposite ends of cylinder

Now hold slide valve nut exactly in centre of space between slide valve lugs, screw valve rod through this nut until knuckle eye is in line with link eye, and push link pin in place. Repeat this process with other side of pump and the operation is complete. It will be found an advisable plan to move both pistons to middle stroke before touching either slide valve.

Before putting on the steam chest covers, move one of the slide valves so as to open the steam port, otherwise the pump might not start, as in setting the valves the steam ports have been covered.

All steam valves are properly set before the pumps leave the factory.

Advantage of Duplex Pumps

The advantage of a duplex pump over other types of direct acting pumps is its continuous discharge. The stroke of one piston begins before the other piston has come completely to rest, so the movement of the suction and discharge columns is practically continuous.

Horizontal Piston Pumps, Simplex Design

Horizontal piston packed pumps are generally built for fluid pressures of 100 to 150 pounds per square inch, although the smaller sizes are suitable for working pressures up to 250 pounds per square inch.

Pumps of the piston packed pattern are built with the suction and discharge valve decks arranged above the piston. Small pumps of this type have the suction and discharge openings on either side as shown in figs. 95a and 95b, or the suction on one side, and discharge on the other, as shown in fig. 100.

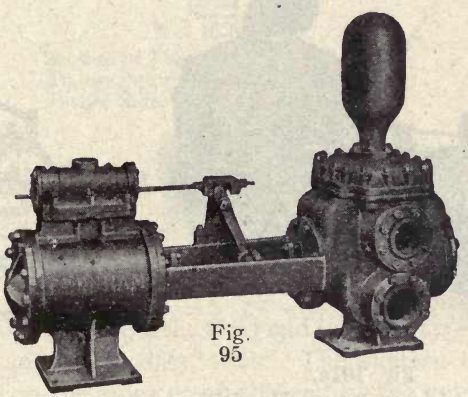


Fig. 95

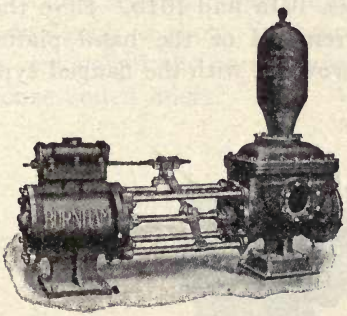


Fig 100
Horizontal Piston Pump with suction one side, discharge the other

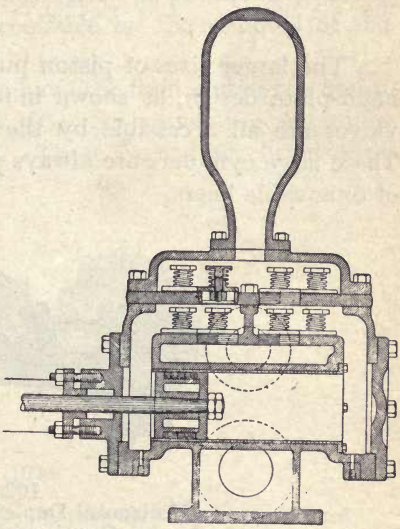


Fig. 95b.
Section of fluid end showing suction and discharge openings either side.

These pumps are fitted with either a pressed bronze liner, or a bolted removable bronze flanged liner.

On these small sizes of piston pumps there is not sufficient room for hand plates, so access to the valves is gained by removing the hood and valve plate.

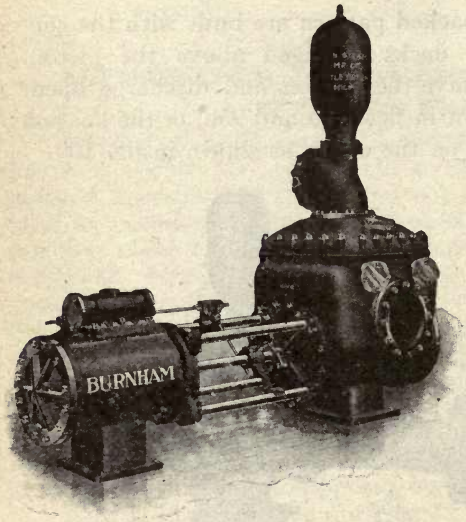


Fig. 101a,
Horizontal Piston Pump, Hand-Plate Design.

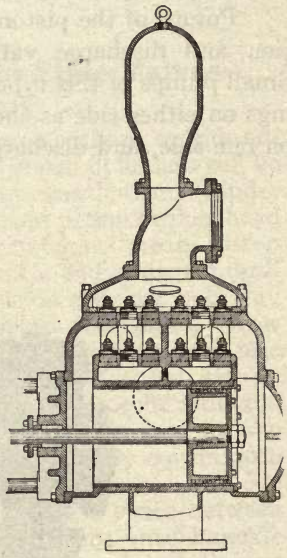
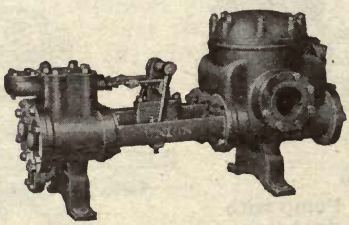


Fig. 101b.
Section of Fluid End, Hand-Plate Design.

The larger sizes of piston pumps have the fluid end of the hand-plate design, as shown in figs. 101a and 101b. Here the valves are all accessible by the removal of the hand plates. These large cylinders are always provided with the flanged type of removable liners.



102a
Horizontal Duplex Piston Pump, Suction Opening on Side, Discharge at End.

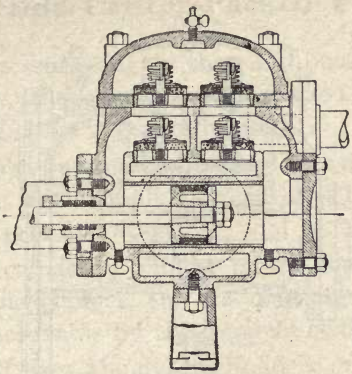


Fig. 102b.
Section of Fluid End of Duplex Piston Pump, Showing Suction Either Side, Discharge at End.

Horizontal Piston Pumps, Duplex Design

Duplex Piston Pumps are built for fluid pressures up to 250 pounds per square inch, the pressure depending on the size of pump.

On small duplex pumps, the fluid end is of the type as shown in figures 102a and 102b, 103a, 103b. The suction opening is arranged either on the sides or at the end of the cylinder, and the discharge is at the end of the cylinder. These cylinders are fitted with bronze liners either bolted or pressed depending on the size. The valves are accessible by the removal of the hood and valve plate.

Large sizes of duplex pumps have the fluid end of the hand-plate design as shown in figure 104. All valves are accessible by removal of the hand plates. These pumps are fitted with bronze bolted liners.

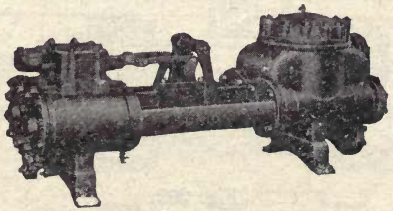


Fig. 103a

Horizontal Duplex Piston Pump, Suction and Discharge at end

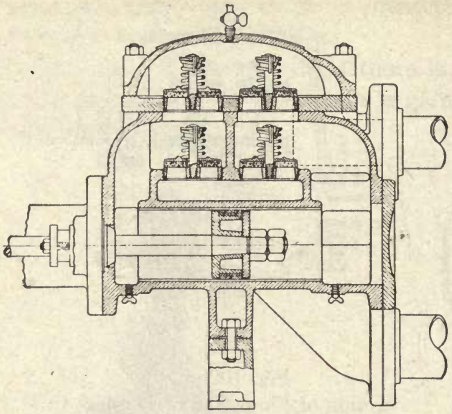


Fig. 103b.

Section of Fluid End of Duplex Piston Pump, showing end Suction and Discharge.

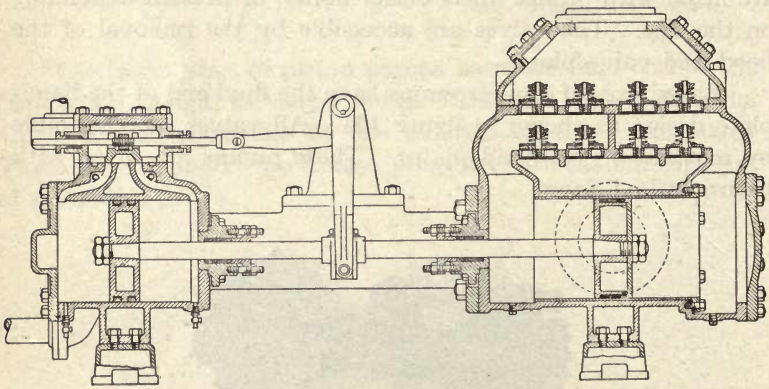


Fig. 202.

Section of Duplex Piston Pump, Hand-Plate Design.

Outside Center-Packed Pumps

For fluid pressures higher than 150 pounds, and up to 300 pounds pressure, the outside center-packed plunger pump, as illustrated in fig. 105a is generally used.

In this type of pump the plunger glands are easily accessible, and any leakage from the plunger can be detected, and stopped while the pump is in operation.

In the center packed plunger pump, the suction valves are located below the plunger, and the discharge valves are above. All valves are accessible by the removal of the hand plates.

In the small sizes, the fluid cylinder is of the type as shown in fig. 105b, while in the larger sizes the cylinders are bolted together as shown in fig 106

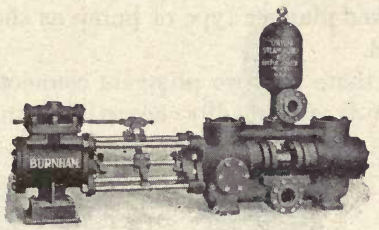


Fig. 105a
Outside Center-Packed Pump.

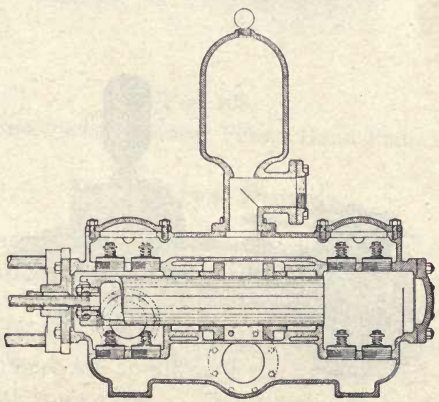


Fig. 105b
Section of Fluid End of Outside Center-Packed Pump, used on smaller sizes.

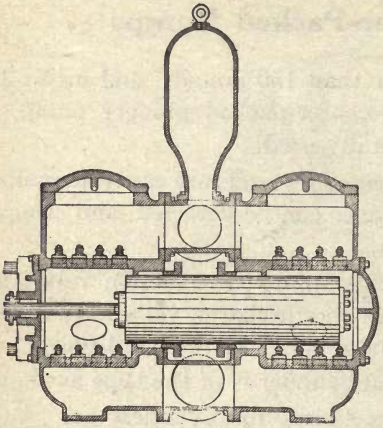


Fig. 106.

Section of Fluid End of Outside Center-Packed Pump, used on large sizes.

Outside End-Packed Plunger Pumps

For fluid pressures up to 150 and 250 pounds per square inch, the end-packed plunger type of pump as shown in fig. 107a is very often used.

In this type there are two plungers connected by side rods. The plunger glands are readily accessible, and any leakage from the plungers can be detected and stopped, while the pump is operating.

The outside end-packed fluid cylinder is of the valve-plate design. In the smaller sizes the valves are accessible by the removal of the hood, and the valve plate fig. 107b. The larger sizes have hand plates for gaining access to the valves as shown in fig. 108.

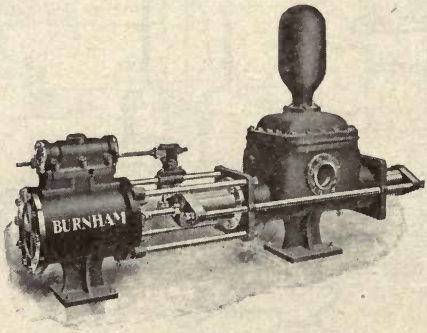


Fig 107a
Outside End-Packed Plunger Pump

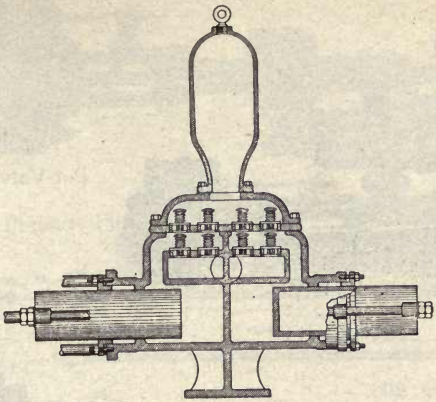


Fig. 107b

Section of Fluid End of Outside End-Packed Plunger Pump, small sizes.

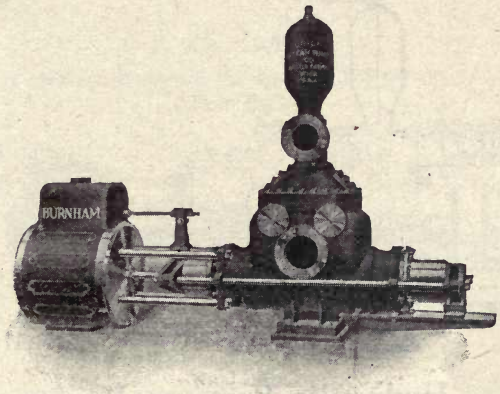


Fig. 108.

Outside End-Packed Plunger Pump, Hand-Plate Design.

Pot Valve Pumps

For fluid pressures of 150 pounds and over, the End-Packed Pot-valve Pump as shown in figs. 109a and 203 is used.

On account of the high pressures this type of pump is subject to, the castings are reduced to the smallest possible dimensions. The valve chambers are small, and the valves used are generally of the bevel-seat wing type. The valves are readily accessible by removing the cover over the valve. See fig. 109 b.

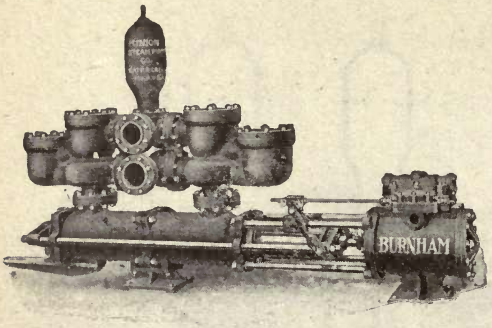
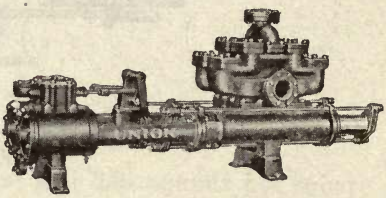


Fig. 109a.
Pot-Valve Plunger Pump

Fig. 203.
Duplex Pot-Valve Plunger Pump.



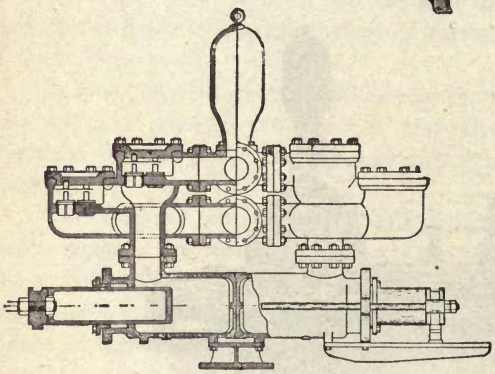


Fig. 109b.
Section of Fluid End of Pot-Valve Pump.

Hydraulic Pressure Pumps

For higher pressures up to 2000 pounds per square inch, the hydraulic pump with cast iron fluid end as shown in figs. 110a and 110b is generally used.

For pressures up to 3000 pounds per square inch, these pumps are fitted with cast steel fluid ends, and for pressures above 3000 pounds, the fluid ends are made of forged steel, as shown in Figs. 111a and 111b.

In the forged steel cylinders all plunger and valve openings are drilled from the solid forging.

In hydraulic pressure pumps the valves used are of the bevel-seat wing type. These valves are accessible by the removal of the screwed plugs over the valves.

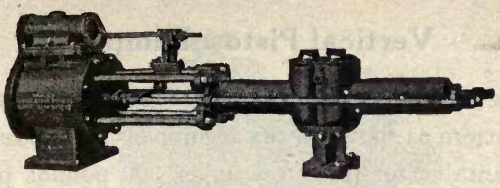


Fig. 110a

Hydraulic Pressure Pump with Cast Iron or Cast Steel Fluid End.

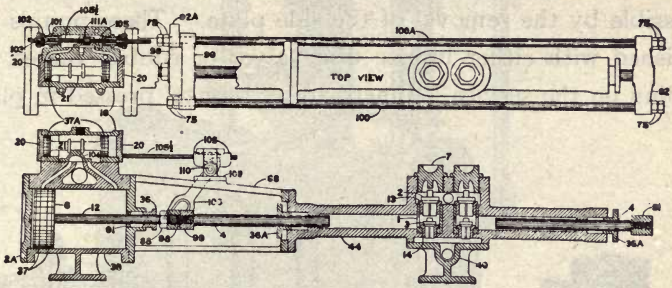


Fig 110b.

Section of Hydraulic Pressure Pump with Cast Iron or Cast Steel Fluid End

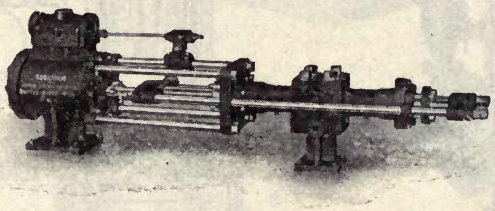


Fig. 111a.

Hydraulic Pressure Pump with Forged Steel Fluid End.

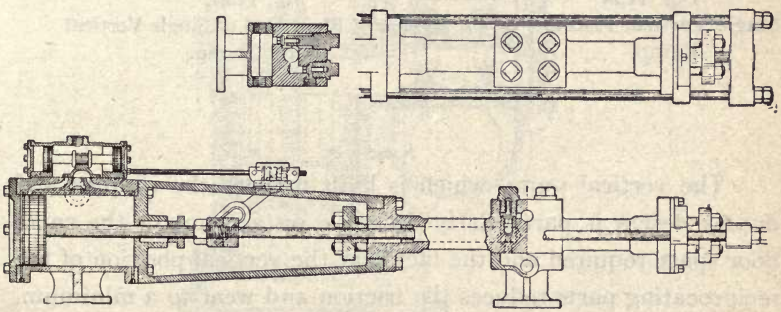


Fig. 111b.

Section of Hydraulic Pressure Pump with Forged Steel Fluid End.

Vertical Piston Pumps

The type of fluid end generally used on piston pumps is of the piston pattern as shown in the accompanying figures. These pumps are suitable for pressures up to 300 pounds per square inch.

The design of the fluid end is such that all valves are easily accessible by the removal of the side plate. These pumps are furnished with either bronze disc valves or rubber valves, depending on the service. Liners are of bronze, pressed in place

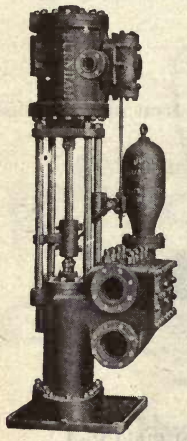


Fig. 112a
Single Vertical Pistor Pump.

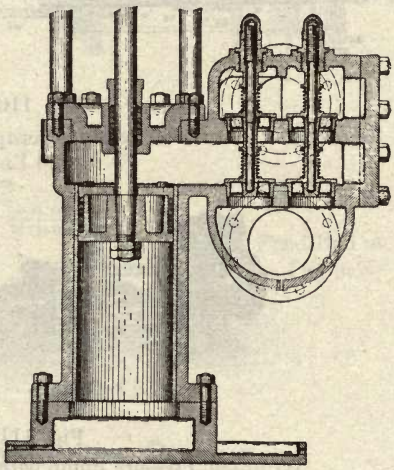


Fig. 112b.
Section of Fluid End of Single Vertical Piston Pump.

The vertical pump which is built in both the simplex and duplex design is particularly desirable on account of the small floor space required and the fact that the vertical position of the reciprocating parts reduces the friction and wear to a minimum.

Vertical pumps are built with a base for floor or deck mounting, or with brackets for bolting to the wall or bulkhead.

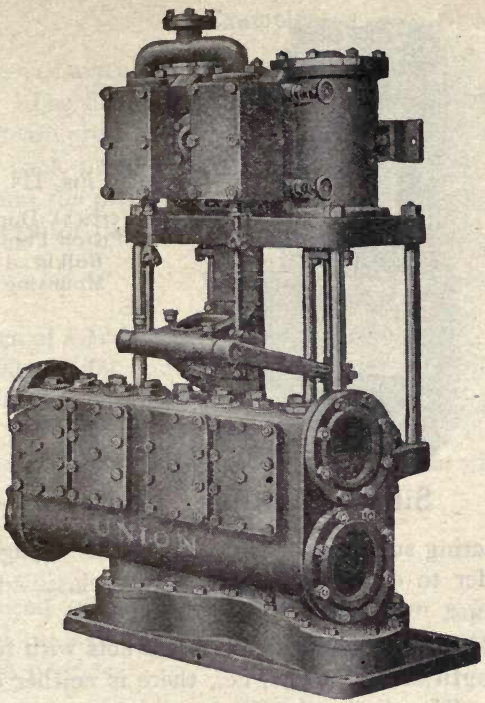


Fig. 113a
Vertical Duplex Piston Pump.

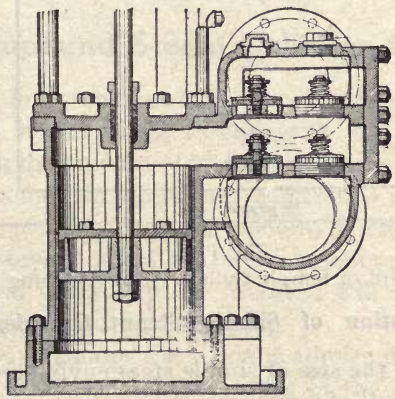


Fig. 113b
Section of Fluid End of Vertical Duplex Piston Pump.

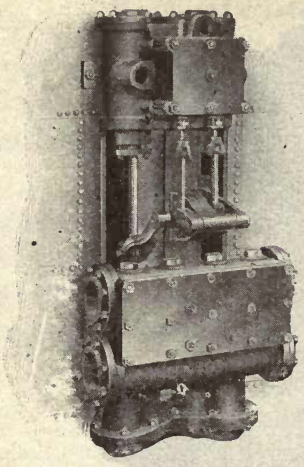


Fig. 114
Vertical Duplex Piston Pump, Bulkhead Mounting

Simple Cylinder Pumps

Direct-acting steam pumps having only one (high pressure) steam cylinder to each fluid cylinder, are classified as simple **cylinder** pumps.

In simple cylinder pumps, the steam acts with full pressure the entire length of the stroke, i.e., there is neither cut off nor compression. The indicator diagram taken from a simple cylinder direct-acting pump is practically rectangular as shown in figure 115.

The mean effective pressure in the steam cylinder of a simple cylinder pump is equivalent to the initial steam pressure per square inch minus the back pressure per square inch.

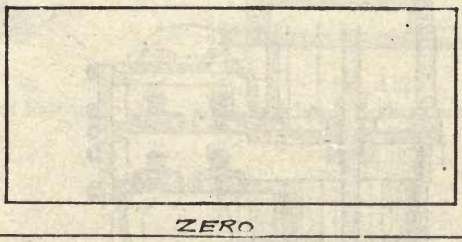


Fig. 115
Indicator Card of Simple Cylinder Direct-Acting Pump.

Calculation of Simple Steam Cylinders

In calculating the size of simple steam cylinders, it is neces sary to know the type of pump, whether piston or outside packed plunger, the diameter of the fluid cylinder, the fluid pressure, the initial steam pressure, and the exhaust or back pressure.

Example

Assume we have a piston type fluid cylinder 12″ in diameter, 16″ stroke. The fluid pressure is 100 pounds per square inch, the initial steam pressure is 110 pounds per square inch, and the exhaust or back pressure is 5 pounds per square inch. It is desired to find the proper size simple steam cylinder, which will be satisfactory for the conditions.

Solution

The area of a 12″ piston = 113 square inches.

113x100 = 11300 pounds total pressure on fluid piston.

Referring to page 295, the mechanical efficiency of a 16 inch stroke piston pump is 80x80 = 64 per cent. Then the total load or pressure to be exerted by the steam piston will be

$$\frac{11300}{.64} = 17656 \text{ pounds.}$$

The initial gauge pressure at the steam cylinder is 110 pounds, and the back pressure 5 pounds, making the net steam pressure 105 pounds.

$$\text{Then } \frac{17656}{105} = 168.1 \text{ Square inches.}$$

= Area of steam piston.

The nearest commercial size of steam cylinder will be 16 inches, and the size of the pump will then be 16×12×16.

Compound Steam Cylinder Pumps

Direct-acting steam pumps take steam during the entire stroke, which makes them extravagant in the use of steam compared with the amount of work done.

To overcome this inherent difficulty, compound steam cylinders are resorted to where economy is essential. In compounding, however, unless the boiler pressure is 80 pounds or over, additional initial expense involved will not be warranted unless the pump operates condensing.

In compound pumps the steam is admitted to the high pressure cylinder during the entire length of the stroke, so that it is not used expansively. When the exhaust port of the high pressure cylinder is opened, the pressure immediately drops to

equalize the pressure on the exhaust side of the high-pressure cylinder, the receiver pipe, and the low-pressure steam chest. When the pistons move on their return stroke, the exhaust pressure in the high-pressure cylinder falls to the pressure in the low-pressure cylinder, this drop being in accordance with Boyle's Law, for the area of the low-pressure cylinder being greater than the high pressure, as the pistons advance, the total volume increases, and the steam expanding reduces in pressure. Figure 116 illustrates the indicator card of a compound direct-acting pump.

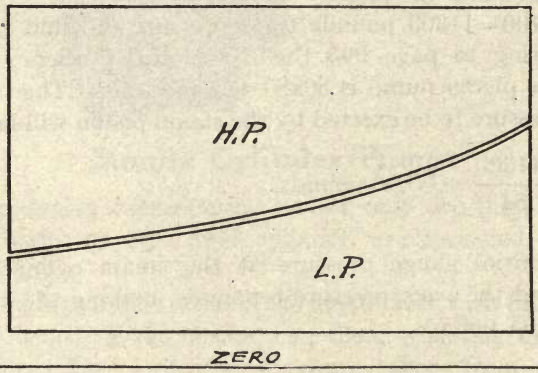


Fig. 116
Indicator Card of Compound Direct-Acting Pump.

Gain in Compounding

The percentage of gain by compounding varies from 25 to 35 per cent in non-condensing pumps, and 25 to 40 per cent in condensing pumps, depending upon the conditions of operation.

Ratio of Cylinders in a Compound Pump

The ratio of cylinders in a compound pump varies from two to three. This ratio is generally dependent on the initial cost, which is based on using standard commercial sizes.

Formulae for Calculating Compound Pumps

The sizes of compound cylinders to use on a direct-acting pump may be calculated by the following formula.

In which I = Initial absolute steam pressure.

B = Absolute back pressure.

Absolute back pressure is 16 pounds for non-condensing pumps, and 6 pounds for condensing pumps.

R = Ratio of steam cylinders.

A = High pressure cylinder area.

Effective pressure, high-pressure cylinder $= I - \frac{I}{R}$ (a)

Effective pressure, low-pressure cylinder $= \frac{I}{R} - B$

Then $R\left(\frac{I}{R} - B\right) = I - RB$ equals (b)

the effective pressure in the low-pressure cylinder referred to the high-pressure cylinder.

The sum of (a) and (b) gives the total effective pressure referred to the area of the high-pressure cylinder or

$$\text{total effective pressure} = I - \frac{I}{R} + I - RB = 2I - RB - \frac{I}{R}$$

Hence the total pressure exerted by the steam cylinders of a compound pump referred to the area of the high-pressure cylinder equals

$$A \times \left(21 - RB - \frac{I}{R}\right) \qquad (47)$$

Example

Assume we have an outside center-packed type fluid cylinder 14 inches in diameter, 20 inch stroke. The fluid pressure is 150 pounds, the initial steam pressure 125 pounds gauge, and the exhaust or back pressure 16 pounds absolute. It is desired to find the proper size compound steam cylinders, which will be satisfactory for the conditions.

Solution

Area of 14 inch plunger = 154 square inches.

$154 \times 150 = 23100$ pounds = total pressure on fluid plunger.

Referring to page 295, the mechanical efficiency of a 20″ stroke outside packed pump is $80 \times 80 = 64$ per cent, then the total load or pressure to be exerted by the steam pistons will be

$$\frac{23100}{.64} = 36093 \text{ pounds.}$$

The initial steam pressure is 125 pounds gauge, or $125 + 14.7$ absolute.

The back pressure is 16 pounds absolute. Now, referring to formula 47, and substituting the initial steam pressure, the back pressure, the total effective pressure, and a cylinder ratio of 3

$$36093 = \text{area high-pressure cylinder} \times \left(2 \times 139.7 - 3 \times 16 - \frac{139.7}{3}\right)$$

Simplifying $\frac{36093}{184.9} = 195$ sq. in. = area of high-pressure cylinder.

This is the approximate area of a 15¾″ cylinder.

The area of the low pressure cylinder will then be $195 \times 3 =$ 585 square inches, or approximately 27″ diameter.

The nearest commercial sizes will be 16 inches high-pressure, and 26 inches low-pressure, so the size of the compound pump in question will be 16 and $26 \times 14 \times 20$.

The following table gives the proper sizes of steam cylinders for compound pumps using steam pressures from 80 to 200 pounds. In this table is given the total pressure exerted by steam cylinders as calculated by formula 47.

The cylinder ratios given are standard commercial sizes, and those recommended for the pressures given.

Compound Steam Cylinders

Total Pressures exerted by various sizes with steam pressures of 80-200 pounds per square inch gauge pressure, and 16 pounds absolute back pressure.

High Press Cylinder	Low Press. Cylinder	R	STEAM PRESSURE 80	90	100	110	120	130	140	150	160	170	180	190	200
8	12	2.25	5596	6378	7160	7941	8723	9505	10287	11069	11850	12633	13415	14197	14979
10	16	2.56	8753	10017	11281	12545	13809	15073	16337	17601	18865	20129	21392	22656	23920
12	18	2.25	12390	14349	16108	17867	19627	21386	23146	24905	26664	28424	30183	31943	33702
12	20	2.78	12541	14397	16252	18106	19962	21817	23672	25527	27383	29238	31093	32948	34803
14	20	2.04	16985	19309	21634	23958	26282	28606	30930	33255	35579	37903	40227	42551	44876
14	22	2.47	17166	19621	22076	24530	26992	29448	31904	34359	36815	39270	41726	44182	46637
14	24	2.94	16955	19509	22064	24619	27173	29727	32284	34839	37393	39948	42508	45063	47618
16	24	2.25	22385	25514	28642	31771	34899	38027	41149	44277	47405	50532	53660	56787	59915
16	26	2.64	22378	25638	28900	32159	35419	38679	41939	45200	48460	51720	54980	58241	61500
18	26	2.09	28157	32028	35900	39772	43645	47516	51388	55260	59132	63004	66876	70748	74620
18	30	2.78	28209	32383	36557	40731	44905	49079	53253	57427	61601	65775	69949	74123	78298
20	30	2.25	34939	39856	44743	49638	54517	59404	64291	69178	74065	78952	83838	88725	93613
22	36	2.67	42280	48459	54639	60818	66998	73178	79357	85537	91716	97896	104076	110255	116435
24	36	2.25	50357	57394	64431	71469	78506	85544	92581	99619	106656	113694	120731	127769	134806
26	42	2.61	59129	67714	76298	84882	93466	102050	110634	119218	127802	136387	144971	153555	162139

Types of Compound Pumps

Compound pumps are built with the high-pressure steam cylinder outboard, as shown in figure 117 or with the low pressure steam cylinder outboard, as shown in figure 118. The latter type, which is known as a three-rod pattern compound is particularly desirable, on account of the accessibility of the pistons and rods.

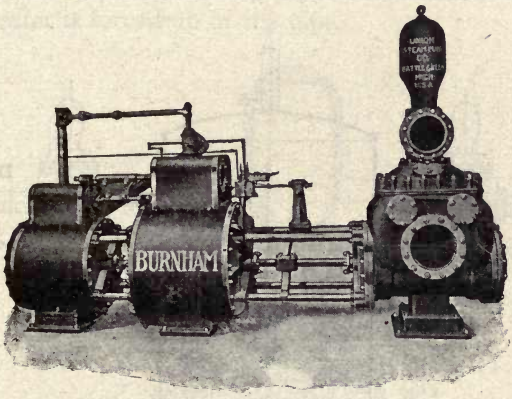


Fig. 117.

Compound Piston Pump.

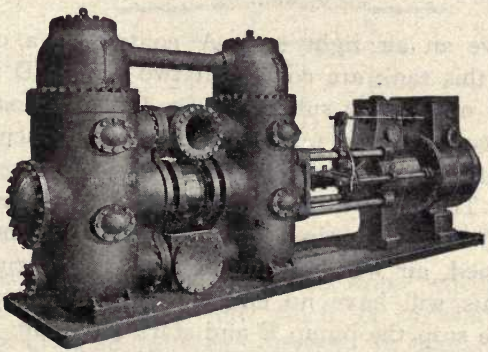


Fig. 118

Compound Center-Packed Plunger Pump.

Suction

Suction is a term used to denote a vacuum, since suction cannot be produced without removing the atmospheric pressure. Elevating water by suction is raising water by means of or through the agency of a vacuum. Simply creating a vacuum, however, will not enable water to be raised. A vacuum in itself is not capable of raising water. We must have pressure to raise or force the water up, and this pressure must be applied on the opposite side of the water to be raised to the space in which the vacuum is created. This is illustrated in figure 119.

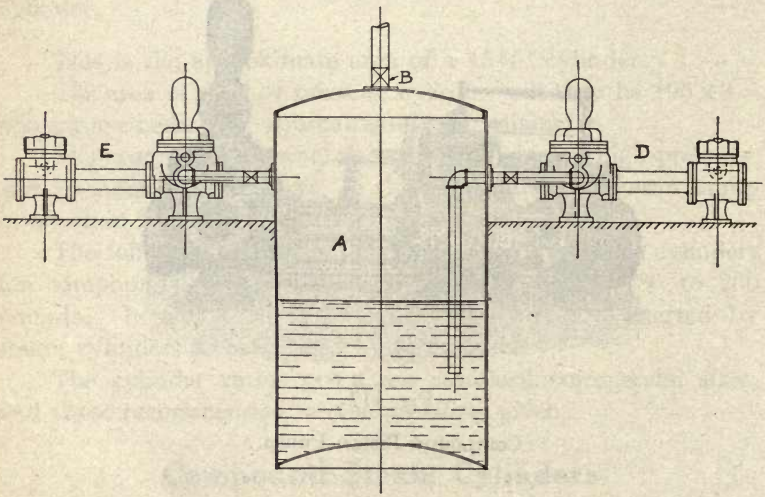


Fig. 119
Sketch illustrating how water is raised by suction.

We have an air tight tank A containing a quantity of water. To this tank are connected two pumps D and E, the suction pipe of D being submerged, and the suction pipe of E not submerged. If the pump E is started, a vacuum will be created in the tank A above the water, and the water will be in a vacuum. The pump cannot raise the water, because there is no pressure to raise it to the pump. If valve B in the top of the tank is opened, air will be admitted, and the vacuum will be broken. This will have no effect upon the water. Assume now that we stop the pump E and start the pump D with the valve B open. A vacuum is created in the suction pipe of pump D above the water, and the pressure on the surface of the water in the tank will force the water up in the pipe of the pump.

These are the principles involved in lifting water by suction. In figure 120 is illustrated a direct acting pump cylinder and suction pipe. As soon as the pump removes a portion of the air pressure inside the pipe, the pressure inside and outside will be unbalanced. As water under pressure presses equally in all directions, if the resistance be removed at any point, the water will flow in that direction, being forced along or upward by reason of the unbalanced pressures. In other words, when a partial vacuum is created in the suction pipe, the pressure or resistance at this point is decreased, and the pressure on the same area of water outside the pipe is, therefore, the greatest, and the water is forced up in the pipe.

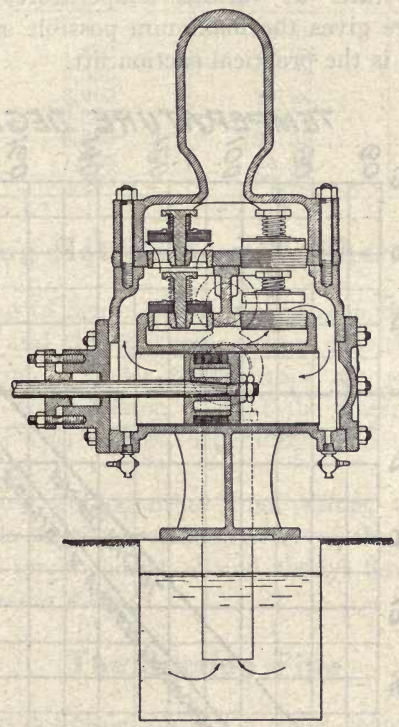


Fig. 120.
Sketch Showing Fluid Cylinder and the Course of Fluid Through Same.

The distance or height to which water will be forced up in the pipe depends upon how much, or to what extent the resistance or pressure has been removed. A column of water 1″ square, and 27.6″ high, weighs 1 lb. If the pressure on each square inch of the surface of the water in the pipe be reduced

1 lb., the water will rise 27.6″ in the pipe. If 2 lbs. pressure be removed, it will rise twice as high or 55.2″, and so on. Now the pressure on the outside of the pipe, i.e., the air pressure down on the surface of the water, is merely the weight or pressure of the atmosphere, which at sea level is 14.7 lbs. on each square inch. It will be seen, therefore, that there is a limit to the height to which the water may be raised by suction, or atmospheric pressure. The maximum theoretical height is equal to 14.7×27.6″, which is 405.72″, or 33.83 feet. Owing to leakage of air and frictional losses, this height is reduced to 26 feet in practice.

The curve below gives the theoretical and practical suction lift of water at various temperatures at sea level. The upper curve gives the maximum possible suction lift, and the lower curve is the practical suction lift.

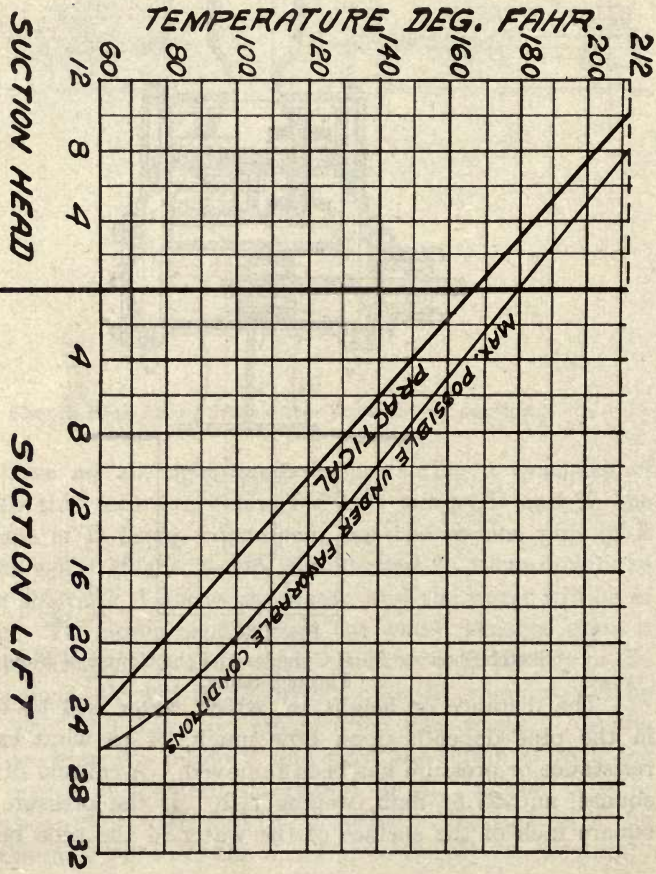


Fig. 121
Suction Lift-Head Curve for Reciprocating Pumps.

Suction at Altitudes

For altitudes above sea level where the barometric pressure is less than 14.7 lbs. per square inch, the water suction lift is less than at sea level. A pump that will raise water 26 feet at sea level would only raise it 21.4 feet if placed at an altitude of one mile. The following table gives the maximum lift possible for different altitudes.

SUCTION LIFT AT DIFFERENT ALTITUDES

Miles	Feet	Theo. Lift.	Actual Lift.
Sea Level	0	33.83	26
¼	1,320	32.38	24.9
½	2,556	30.79	23.7
¾	3,960	29.24	22.5
1	5,280	27.76	21.4
1¼	6,600	26.38	20.3
1½	7,836	25.13	19.3
2	10,560	22.82	17.5

Handling Hot Water and Other Liquids

By referring to figure 121, it will be seen that for handling hot water 168° and above, it is necessary that the water should flow to the pump. Boiler feed pumps are generally placed below the feed water heater, so that the water flows to them under a head of 5 to 10 feet, depending upon the temperature of the feed water.

For pumping liquids other than water, the suction lift possible depends on the specific gravity of the liquids. Thick liquids, such as tar, molasses, should always flow to the pump by gravity, or under a head.

The Suction Pipe

It is always desirable to place a pump as near the source of supply as possible. While it requires no more power to raise water by suction than to discharge it an equal distance, the disadvantage of the long suction line is that a very small leak under a high vacuum is likely to disable the pump, while a leak several times the size in the discharge line would not affect the pump. Furthermore, owing to the larger size of the suction line, a long line adds to the cost of the installation.

The height of lift is always measured vertically between the surface of the water, and the center of the suction inlet at the pump. If the water level is liable to fluctuate, the height at which the pump is placed above the water is reckoned from the lowest water level.

With long suction lines, the friction loss should be taken into consideration, when calculating the suction lift.

When a pump is to operate on a suction-lift of 15 feet or more, it is advisable to install a foot valve on the suction pipe in order to insure a smooth operation of the pump. The object of a foot valve is to keep the pump primed, and it allows the pump to start off easily, without having to fill the suction pipe at each stroke.

In freezing weather, provision must be made for draining the suction pipe and water cylinder.

The suction pipe should be carefully laid, and all joints made tight. The suction line should be securely suspended to avoid any vibration, which might cause a leak. Where the suction pipe is very long, or the lift high, a suction air chamber on the suction pipe assists the pump in starting the long column of water at each stroke, and it also stops the motion of the water without shock, in case the pump is operated at high speed. Suction air chambers are preferably placed as shown in figure 122.

Size of Suction and Discharge Pipes

If the speed of the water in the suction pipe were to be kept the same as the speed of the pump piston or plunger, the suction pipe of course would have to be the same size as the water cylinder. While this would represent an ideal condition as far as friction is concerned, it would also mean a very large suction pipe, and an unnecessary added expense. It is considered good practice to strike a mean between the loss by water friction and the use of an exceptionally large suction pipe. In doing this it has been found from experience that a velocity of approximately 250 feet per minute in the suction pipe will give good results. The area of the suction pipe, therefore, will be as much smaller than the area of the cylinder, as the piston speed is less than 250 feet per minute. If the piston speed was 100 feet per minute, then the area of the suction pipe would be 40% of the area of the water cylinder, or cylinders. In

other words, the area of the suction pipe in square inches is equal to the area of the water piston times the piston speed in feet per minute, divided by 250.

Example

Calculate the size of the suction pipe for a 12×8×12 simplex pump based on the pump making 100 strokes per minute, or 100 feet piston travel.

Solution

The area of the water cylinder or piston is 50.26 square inches, and the piston speed in feet per minute is 100. The area of the suction pipe will then be

$$\frac{50 \times 100}{250} = 20.1 \text{ Square inches.}$$

which corresponds to a diameter of about 5″.

The size of the discharge pipe may be calculated in the same manner by using a velocity of 400 feet per minute through the discharge pipe, instead of 250 feet.

Example

Calculate the size of the discharge pipe for a 12x8x12 pump based on the pump making 100 strokes per minute or 100 feet piston travel.

Solution

The area of the 8″ water cylinder or piston is 50.26 square inches, and the piston speed in feet per minute is 100. The area of the discharge pipe will then be

$$\frac{50.26 \times 100}{400} = 12.56 \text{ Square inches.}$$

which corresponds to a diameter of 4″.

Assuming a practical velocity of water in the suction pipe of 250 feet per minute, and a velocity in the discharge pipe of 400 feet per minute, the sizes of suction and discharge pipes may be calculated from the following formulae:

$$D_1 = \sqrt{.097\ G} \qquad (48)$$

$$D_2 = \sqrt{.0612\ G} \qquad (49)$$

In which D_1 = Diameter of suction pipe.

D_2 = Diameter of discharge pipe.

G = Gallons per minute.

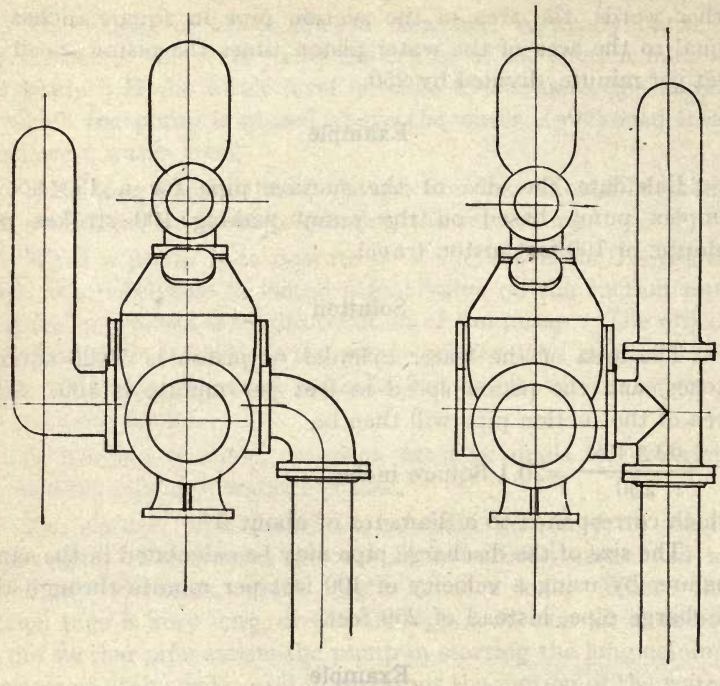


Fig. 122
Sketch Showing Proper Location of Suction Air Chamber.

Air Chambers

Air chambers are generally of cast iron, and may be attached to the discharge pipe or to the suction pipe. When the air chamber is attached to the discharge pipe, it is called a discharge air chamber, and when attached to the suction pipe, it is called a suction air chamber. The object of an air chamber is to provide an elastic element in the pipe line to take up the shocks and pulsations, and produce a uniform flow in the pipe. The discharge air chambers are generally furnished on simplex pumps of all kinds, except those used for vacuum service. The discharge air chambers are not furnished on the smaller sizes of duplex pumps. However they are advisable on larger sizes.

Discharge air chambers are generally provided with an opening on the side near the top for charging with air. However if there is not a supply of compressed air available, a snifting valve of pet cock may be placed in the suction, which will admit sufficient air to the water to properly charge the air chamber.

Where the suction pipe is very long, or the suction lift high, an air chamber on the suction pipe is recommended, as it assists the pump in starting the long column of water at each stroke, and it also stops the motion of the water without shock, in case the pump is operated at high speed. Suction air chambers should be placed as shown in figure 122, and the capacity of the suction air chamber should be approximately 6 to 8 times the capacity of the water cylinder.

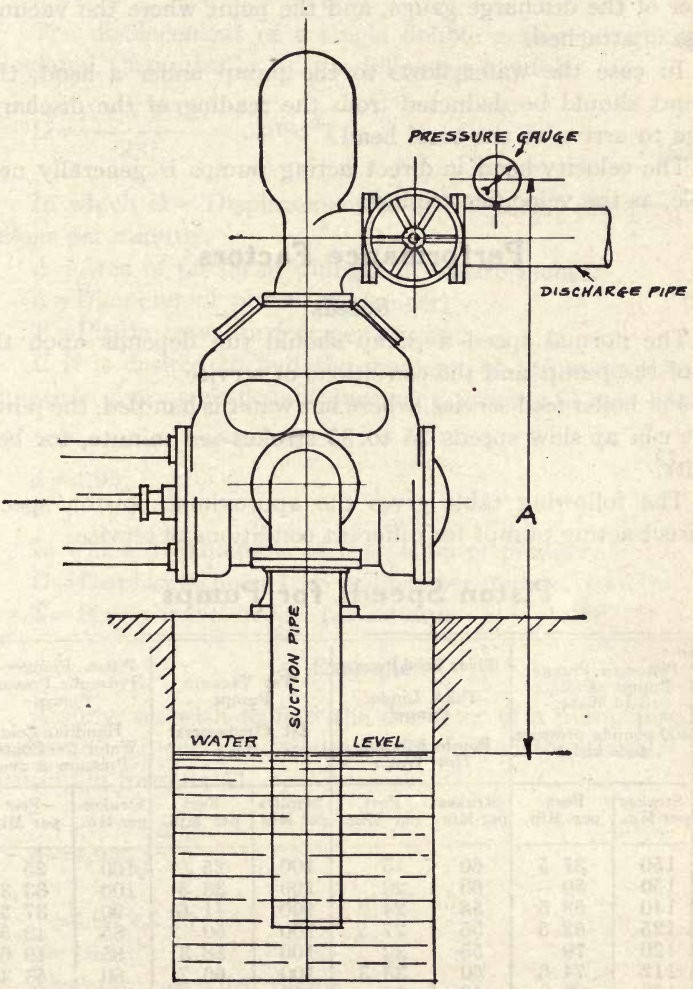


Fig. 123

Sketch Showing Arrangement of Piping on Water End to Determine the Total Head.

Measurement of Total Head

The total head of a pump may be found by a test gauge placed on the discharge pipe, and to its reading must be added the distance from the center of the gauge to the level of the water in the suction well (see figure 123). If the suction pipe is long, the suction lift may be found by placing the vacuum gauge on the suction pipe close to the pump. The readings of the two gauges are added, as is also the distance between the center of the discharge gauge, and the point where the vacuum gauge is attached.

In case the water flows to the pump under a head, this amount should be deducted from the reading of the discharge gauge to arrive at the total head.

The velocity head in direct acting pumps is generally negligible, as the velocities are low.

Performance Factors

Speeds

The normal speed a pump should run depends upon the size of the pump and the conditions of service.

For boiler feed service, where hot water is handled, the pump must run at slow speeds 25 to 35 strokes per minute, for best results.

The following table gives the approximate piston speeds of direct acting pumps for different conditions of service.

Piston Speeds for Pumps

Stroke of Pump	Piston or Plunger Pumps handling Cold Water 100 pounds pressure and under		Boiler Feed Pumps Thick Liquor Pumps Pumps handling Hot Water		Wet Vacuum Pumps Jet Condensers		Piston, Plunger or Hydraulic Pressure Pumps Handling Cold Water 150 Pounds Pressure or over	
	Strokes per Min.	Feet per Min.	Strokes per Min.	Feet per Min.	Strokes per Min.	Feet per Min.	Strokes per Min.	Feet per Min.
3	150	37.5	60	15	100	25	100	25
4	150	50	60	20	100	33.3	100	33.3
5	140	58.5	58	24.2	100	41.6	90	37.5
6	125	62.5	55	27.5	100	50	85	42.5
7	120	70	55	32	100	58.3	85	49.6
8	112	74.6	50	33.3	100	66.7	80	53.3
10	100	83.3	42	35	100	83.3	75	62.5
12	90	90	35	35	100	100	75	75
16	75	100	35	46.7	75	100	60	80
20	60	100	30	50	60	100	54	90
24	50	100	25	50	50	100	45	90

For cold water service against low pressures, 100 pounds and under, piston speeds up to 100 feet per minute depending upon the size of the pump, can be successfully used.

For high pressure service, 150 pounds and over, pumps are generally operated at moderate speed.

Pumps handling thick liquids like syrup, molasses, heavy oil, etc., which cannot be made to flow fast should also run slow.

Displacement

The displacement of a single double acting pump can be calculated theoretically by the following formula:

$$D = \frac{A \times T \times 12}{231} = .0408 d^2 T \qquad (50)$$

In which D = Displacement of double acting plunger, U. S. gallons per minute.

A = Area of piston or plunger in square inches.

d = Diameter of piston or plunger.

T = Piston travel in feet per minute.

If it is desired to find the diameter of the water piston or plunger to give a specified displacement, formula 51 may be used.

$$d = 4.95 \sqrt{\frac{D}{T}} \qquad (51)$$

In which d = Diameter of the piston or plunger.

D = Displacement in U. S. gallons per minute.

T = Piston travel in feet per minute.

Example

Assume we wish to find the diameter of a pump piston to handle 800 gallons per minute at 75 feet piston travel. Substituting in formula 51.

$$d = 4.95 \sqrt{\frac{800}{75}}$$

$$= 4.95 \times 3.26$$

$$= 16.13''$$

Therefore use a 16″ piston.

In the center packed plunger, or the piston pump, the displacement of the piston rod must be deducted, if accurate results are desired.

Slip

There is a loss of capacity in the operation of a pump due to leaky valves, piston packing, stuffing boxes, or suction. This loss is generally stated as a percentage of the displacement, and is called the slip.

$$\text{Slip} = \frac{L}{D} \qquad (52)$$

In which L = Loss by leakage in gallons per minute.
D = Displacement in gallons per minute.

Slip varies in pumps from 2% to 10%, depending entirely upon the condition of the pump. It is customary to use a figure of 5% in estimating the slip of a pump.

Capacity

The capacity of a pump is the actual volume of liquid delivered, and equals the displacement minus the slip.

$$C = D - S \qquad (53)$$

In which C = Capacity in gallons per minute.
D = Displacement in gallons per minute.
S = Slip in gallons per minute.

The capacity of a pump may be found by calculation, assuming a factor for the slip, or by actually measuring the water by weir, tank, nozzle, or pitot tube, as described on page 110.

In calculating the displacement, or capacity of a pump by the formula given, the results obtained are in U. S. gallons of 231 cubic inches.

In Great Britain, and her colonies, the Imperial gallon is used, which contains 277 cubic inches, or is 20% larger than the U. S. gallon.

Volumetric Efficiency

The volumetric efficiency of a pump is the ratio of the capacity to the displacement, and equals

$$E_v = \frac{C}{D} \qquad (54)$$

Hydraulic Efficiency

The hydraulic efficiency is the ratio of the total head pumped against to the total head pumped against plus the hydraulic losses, equals

$$E_{hy} = \frac{H}{H+L} \qquad (55)$$

The hydraulic losses consist of the frictional losses, in the suction pipe, through the pump valves, and seats, as well as the velocity head.

Mechanical Efficiency

The ratio of the indicated horse power of the water end to the indicated horse power of the steam end is the mechanical efficiency, and equals

$$E_m = \frac{H\,P_w}{HP_s} \qquad (56)$$

The mechanical efficiency of direct acting pumps varies with the size and type from 50% to 90%. This factor can be determined only by actual test.

The following table gives an approximate idea of the mechanical efficiency of direct acting pumps of the piston and outside packed types.

In calculating pump sizes, to take care of any possible drop in steam pressure, or unforeseen conditions, we recommend that the mechanical efficiency be taken at 80% of the values given in the following table.

MECHANICAL EFFICIENCY PERCENT

Stroke of Pump Inches	Piston Type Percent	Outside Packed Plunger Type Percent
3	55	50
5	60	56
6	65	61
7	68	64
8	72	68
10	76	72
12	78	75
16	80	77
20	83	80
24	85	82

Overall Efficiency

This efficiency comprises all losses in the pipe, and indicates the economy of the whole unit. It is expressed as follows:

$$E_o = E_{hy} \times E_v \times E_m \qquad (57)$$

Steam Indicated Horse Power

The steam indicated horse power of a pump is calculated by the formula

$$\text{Horse Power} = \frac{P.\ L.\ A.\ N.}{33000} \qquad (58)$$

P = Indicated M. E. P. pounds per square inch.

L = Length of stroke in feet.

A = Area of steam piston in square inches.

N = Number of strokes per minute.

Water Horse Power

The horse power of the water end equals.

$$H\ P_w = \frac{G \times W \times H}{33000} = .000252\ G.\ H. \qquad (59)$$

Where G = U. S. Gallons per minute delivered.

W = Weight of one U. S. gallon in pounds.

H = Total head pumped against in feet.

Example

Find the water horse power of a pump delivering 500 U. S. gallons per minute against a head of 200 lbs.

Solution

1 lb = 2.31 feet.

200 lbs = 200 × 2.31 = 462 feet.

Now substituting in formula 59,

$H\ P_w = .000252 \times 500 \times 462 = 58.2$

Materials and Manner of Fitting Pumps for Handling Different Liquids and Gases

Direct acting pumps, power pumps, and crank and flywheel pumps are classified according to the manner in which the liquid cylinder is fitted, as follows:

Standard Fitted: This means that the pump is fitted with cast iron liquid cylinder, steel piston rod, bronze liner, bronze valve seats, bolts and springs, rubber or bronze valves, and cast iron liquid piston or plunger.

Standard Bronze Fitted: This signifies a pump fitted with cast iron liquid cylinder, bronze piston rod, bronze liner, bronze valve seats, bolts and springs, rubber or bronze valves, and cast iron liquid piston or plunger.

Full Bronze Fitted: This is a pump, which carries a cast iron liquid cylinder, bronze liner, bronze valve seats, bolts and springs, rubber or bronze valves, bronze liquid piston or plunger.

All Iron Fitted: Pumps for handling tar, ammonia, etc. are built without bronze fittings. In this type of pump the liquid cylinder is of cast iron, the piston rod is of steel, valve seats and valves of cast iron, valve bolts of steel, valve springs of steel, and liquid piston or plunger of cast iron. Iron fitted pumps are furnished with, or without cast iron liners, depending on the size of the pump.

All Bronze: This is a pump with the liquid cylinder of bronze, bronze liner, bronze piston rod, bronze valve seats, bolts and springs, bronze valves and bronze water piston or plunger.

In the above classification, the term "Standard Fitted" generally applies to duplex pumps only. Simplex Pumps are built regularly, "Standard Bronze Fitted."

The following list gives an idea of the proper fitting for pumps for handling different liquids, as well as the type of liquid valve to use, and the kind of packing in the liquid piston or plunger.

The Materials and Fittings Used for Pumping Various Liquids

Direct Acting Pumps, Power Pumps Crank and Flywheel Pumps

The following table gives the proper materials and fittings to be used on pumps for handling different kinds of liquids.

Kind of Liquid	Material Used	Valves	Piston Pack g
Acetic Acid Concentrated	All Bronze	Bronze disc	Hydraulic
Acetic Acid Diluted	All Bronze	Bronze disc	Hydraulic
Acid Mine Water	All Bronze	Bronze disc	Hydraulic
Alkaline Water	All Iron Fitted	Iron disc	Hydraulic
Alcohol (crude)	Standard Bronze fitted	Bronze disc	Hemp
Ammonia Water(Aqua Am.)	All Iron fitted	Iron disc	Hydraulic
Anilin Water	All Iron fitted	Iron disc	Hydraulic
Benzene	All Iron fitted	Iron disc	Hemp
Benzine	Standard Bronze fitted	Bronze disc	Hemp
Beer	All Bronze fitted	Bronze disc	Hydraulic
Beer-wort	All Bronze fitted	Bronze disc	Hydraulic
Beet Juice (thin)	All Iron fitted	Iron disc	Hydraulic
Bisulphite	Standard Bronze fitted	Bronze disc	Hemp
Bitter Mineral Water	All Bronze	Bronze disc	Hydraulic
Brine (Calcium)	All Iron fitted	Iron disc or rubber	Hydraulic
Brine (Sodium)	Standard Bronze fitted	Bronze disc or rubber	Hydraulic
Cane Juice	Full Bronze fitted	Bronze disc	Hydraulic
Carbonate of Soda	All Iron fitted	Iron disc	Hydraulic
Carbonic Acid	Standard Bronze fitted	Bronze disc	Iron Ring
Caustic Carbonate of Soda in Solution (Hot)	All Iron fitted	Iron disc	Hemp
Caustic Chloride of Magnesium Solution (Cold)	All Iron fitted	Iron disc	Hemp
Caustic Cyanogen in Solution	All Iron fitted	Iron disc	Hemp
Caustic Manganese in Solution	All Iron fitted	Iron disc	Hemp
Caustic Potash Solution	All Iron fitted	Iron disc	Hemp
Caustic Potash Niter in Solution	All Iron fitted	Iron disc	Hemp
Caustic Soda Solution	All Iron fitted	Iron disc	Hemp
Caustic Sodium Chloride Solution	All Iron fitted	Iron disc	Hemp
Caustic Zinc Chloride	All Iron fitted	Iron disc	Hemp
Chlorine	Nickel Manganese or Alloy		
Chloride of Potash Solution	All Iron fitted	Iron disc	Hemp
Caustic Chloride of Magnesium Solution (Hot)	Hard lead		
Coal Tar Oil	All Iron fitted	Iron disc	Iron Ring
Creosote Oil	All Iron fitted	Iron disc	Iron ring
Cresol	All Iron fitted	Iron disc	Iron ring
Cyanogen	Standard Bronze fitted	Bronze disc	Hemp
Cyanide of Potassium	All Iron fitted	Iron disc	Hemp
Distillery-wort	All Bronze	Bronze disc	Hemp
Ferrous Chloride	All Iron fitted	Iron disc	Hemp
Gasoline	Standard Bronze fitted or standard fitted	Bronze disc	Hemp or iron ring packing
Glue (Hot)	Full Bronze fitted	Bronze ball	Bronze ring
Glycerin	All Bronze	Bronze disc	Bronze ring
Grease (Hot)	Standard Bronze fitted	Bronze disc	Bronze ring
Green Vitriol	All Iron fitted	Iron disc	Iron ring
Guncotton Brine	All Iron fitted	Iron disc	Iron ring
Hot Oil (300 °.) Max. temp.	Standard Bronze fitted	Bronze disc	Bronze ring
Hot Oil(over 300°.)	All Iron fitted	Iron disc	Iron ring Asbestos in pis ton rod Stuffing box
Heavy Oil	Standard Bronze fitted	Bronze disc	Bronze ring
Hydrochloric acid in thin solution	All Bronze	Bronze disc	Bronze ring
Hydrochloric acid	All Bronze	Bronze disc	Bronze ring
Iron Pyritic Acid	All Bronze	Bronze disc	Bronze ring
Lard (Hot)	All Iron fitted	Iron ball valves	Iron ring

The Materials and Fittings Used for Pumping Various Liquids—Continued

Direct Acting Pumps, Power Pumps, Crank and Flywheel Pumps

The following table gives the proper materials and fittings to be used on pumps for handling different kinds of liquids.

Kind of Liquid	Material Used	Valves	Piston Packing
Lime Water	All iron fitted	Iron disc	Hemp
Linseed Oil	Standard bronze fitted	Bronze disc	Hemp
Lye (Caustic)	All iron fitted	Iron Disc	Iron ring
Lye (containing much salt)	Standard bronze fitted	Bronze disc	Hemp
Lye Solution (containing sand,)	All iron fitted	Iron disc	Hemp
Mash	All Bronze	Bronze ball	Hemp
Milk	All Bronze	Bronze disc or clapper	Solid bronze piston
Mineral Oil	Standard bronze fitted	Bronze disc	Bronze ring
Molasses	Full bronze fitted	Bronze ball	Bronze ring or Hemp
Naphtha	Standard bronze fitted or Standard fitted	Bronze disc	Iron or Bronze ring
Nitric Acid (Concentrated)	Lead		
Nitric Acid (Diluted)	All iron fitted	Iron disc	Hemp
Olive Oil	Standard bronze fitted	Bronze disc	Bronze ring
Paraffin (Hot)	Standard bronze fitted	Bronze ball	Bronze ring
Petroleum	All iron fitted	Iron ball	Iron ring
Petroleum Ether	All iron fitted	Iron disc	Hemp
Pitch (Hot)	All iron fitted	Iron ball	Iron ring
Potash Solution	All iron fitted	Iron disc	Hemp
Pulp	Standard bronze fitted	Special ball valve pump	
Purifying Oil	All iron fitted	Iron disc	Iron ring
Rape Oil	Standard bronze fitted	Bronze disc	Bronze ring
Rosin (Hot)	All iron fitted	Iron disc	Iron ring
Salt Water	Full bronze fitted	Bronze disc or rubber	Hydraulic
Sea Water	Full bronze fitted	Bronze disc or rubber	Hydraulic
Sewage	Full bronze fitted	Bronze disc or rubber	Hydraulic
Sebacic Acid	All bronze	Bronze disc	Hemp
Syrup	Standard bronze fitted	Bronze disc	Bronze ring
Soap Water	All iron fitted	Iron disc	Hemp
Soda	All iron fitted	Iron disc	Hemp
Sodium Chloride Solution	Standard bronze fitted	Bronze disc	Hemp
Sodium Sulphate	All iron fitted	Iron disc	Hemp
Stearic Acid (Hot)	All bronze	Bronze disc	Hemp
Sugar Compound	Standard bronze fitted	Bronze disc	Bronze ring
Sugar Solution	Standard bronze fitted	Bronze disc	Bronze ring
Sulphate of Lime	Standard bronze fitted	Bronze disc	Bronze ring
Strontia in Caustic Solution	All iron fitted	Iron disc	Iron ring
Sulphide of Hydrogen	All bronze	Bronze disc	Hemp
Sulphite of Sodium (Hot)	All iron fitted	Iron disc	Iron ring
Sulphuric Acid Concentrated	All iron fitted	Iron disc	Hemp
Sulphuric Acid Common	All bronze	Bronze disc	Bronze ring
Sulphurous Acid Concentrated	All bronze	Bronze disc	Hemp
Sulphurous Acid, Diluted	All bronze	Bronze disc	Hemp
Tar (Hot)	All iron fitted	Iron ball	Iron ring
Tannic Acid	All bronze	Bronze disc	Hemp
Toluol	Standard bronze fitted	Bronze disc	Hemp
Turpentine Oil	All iron fitted	Iron disc	Iron ring
Vegetable Oil	All iron fitted	Iron disc	Iron ring
Vinegar	All bronze	Bronze disc	Hemp
Wine	All bronze	Bronze disc	Bronze ring
Wood Alcohol	Standard bronze fitted	Bronze disc	Hemp
Water (Hot or Cold)	Standard bronze fitted	Rubber or bronze disc	Hydraulic
Water containing sulphur	Standard bronze fitted	Bronze disc	Hydraulic
Water containing some tar and ammonia	All iron fitted	Iron disc	Iron ring
Wood Pulp	Standard bronze fitted	Special bronze Ball valve pump	
Whisky	All bronze	Bronze disc	Hemp

Duty

Duty is the number of foot pounds of useful work done by 1000 pounds of dry steam, or a 1,000,000 B. T. U.

Duty formerly meant the amount of work done by 100 pounds of coal, but owing to the variation in the quality of the coal, and the efficiency of the boiler, this definition was discarded.

An efficient boiler will evaporate ten pounds of water per pound of coal. Therefore, in order to make a fair comparison $10 \times 100 = 1000$ pounds of steam was adopted as a basis for rating the economy of pumps.

The method of calculating duty is as follows:

Let W = Number of pounds of steam used per horse power per hour.

D = Duty in foot pounds per 1000 pounds of dry steam.

H = Number horse power developed per 1000 pounds dry steam per hour.

Then $\frac{1000}{W} = H.$

Now one horse power $= 60 \times 33000$ foot pounds per hour.

Therefore 1000 pounds dry steam will deliver $H \times 60 \times 33000$ foot pounds per hour.

Or substituting $\frac{1000 \times 60 \times 33000}{W}$ foot pounds per hour.

By definition, the above expression is the Duty D. Therefore writing it down equal to D, and multiplying we have,

$$D = \frac{1{,}980{,}000{,}000}{W} \qquad (60)$$

This formula enables one to readily calculate the rating in feed water or steam consumption, if the duty is given, or vice-versa. Thus a duty of 100 million foot pounds per 1000 pounds of steam is equivalent to 19.8 pounds of steam per horse power hour.

The A. S. M. E. in 1891 reported a standard method of conducting duty, instead of the above units of 100 pounds of coal, or 1000 pounds of steam, they recommended a new unit based on 1,000,000 B. T. U. furnished by the boiler. The economy is then expressed in foot pounds of work done per 1,000,000 B.T.U., this unit is the equivalent of 100 pounds of coal when each pound imparts 10,000 B. T. U. to the water in the boiler.

Measure of duty; the principal data required for determining the duty of a pump is the work done, and the steam consumed in doing this work.

Capacity: The actual amount of water delivered by a direct acting pump may be measured by pitot tube, calibrated tank, nozzle, or Venturi meter, as illustrated and described on page110. However as the displacement pump is really a meter, it is reasonably accurate to calculate the displacement of the plunger and deduct the piston rod, as well as deducting 5% to 10% for slip. The amount of slip depends upon the size of the pump, as well as the conditions of the valves and packing.

Head: The head pumped against is obtained by placing a calibrated pressure gauge on the discharge pipe, and to its reading must be added the difference in the velocity head in the suction and discharge pipes, and the vertical distance from the center of the gauge to the level of the water in the suction well. See figure 123. If the suction pipe is long, a calibrated vacuum gauge should be placed on the suction pipe close to the pump. Then to obtain the total head, the readings of the vacuum and discharge gauges are added, as is also the distance between the center of the discharge gauge and the point where the vacuum gauge is connected, and the difference in the velocity heads in the suction and discharge pipes. The velocity head may be calculated by formula 22, page 108. If the suction and discharge pipes are the same size, the velocity head is zero.

Steam Consumption: To determine the quantity of steam used by the pump, there are two methods employed. The first method is by measuring the amount of feed water pumped to the boiler, and the second method is by measuring the condensate discharged by the air pump. The first method is used where the steam is condensed in a jet condenser in which the steam and injection water are mixed, and the second method is employed where a surface condenser is used.

Example

A compound steam pump pumps 3,000,000 gallons of water per twenty-four hours against a head of 100 pounds per square inch. The steam consumption is 40 pounds per water horse power per hour. Steam pressure 100 pounds, exhaust temperature 120° (a) what is the duty per 1000 pounds of steam? (b) what is the duty per 1,000,000 B. T.U.? A gallon of water weighs 8⅓ pounds, and one pound water pressure equals a head of 2.31 feet.

Solution (a)

Weight of water pumped in 24 hours $3{,}000{,}000 \times 8\frac{1}{3}$
$= 25{,}000{,}000$ lbs.

Head pumped against $= 100 \times 2.31 = 231$ feet.

Work done in 24 hours $= 25{,}000{,}000 \times 231$
$= 5{,}775{,}000{,}000$ foot lbs.

Work done per hour $= \dfrac{5{,}775{,}000{,}000}{24}$
$= 240{,}625{,}000$ foot lbs

Water horsepower $= \dfrac{240{,}625{,}000}{33{,}000 \times 60}$
$= 121.5$

Steam used per hour $= 4860$ lbs.

Duty per 1000 lbs. of steam $= \dfrac{240{,}625{,}000}{4.86}$
$= 49{,}511{,}300$ foot lbs

Solution (b)

Net heat supplied to pump per pound of steam.
$= 1188.6 - (120-32) = 1100.6$ B.T.U. (See steam table in Appendix.)

Total heat furnished to pump by boiler $= 1100.6 \times 4860.$
$= 5{,}348{,}916$, B. T. U.

Duty per 1,000,000 B. T. U. $= \dfrac{240{,}625{,}000}{5.348} = 44{,}993{,}455$ foot pounds

Duty of Pumping Engines

Pounds Steam H. P. Hour	Duty per 1000 Pounds of Steam	Duty per 1000 pounds of coal, on basis of evaporation as follows:				
		9.5 to 1	9 to 1	8.5 to 1	8 to 1	7 5 to 1
100	19,800,000	18,810,000	17,820,000	16,830,000	15,840,000	14,850,000
95	20,842,105	19,799,999	18,757,894	17,715,799	16,673,684	15,631,578
90	22,000,000	20,900,000	19,800,000	18,700,000	17,600,000	16,500,000
85	23,294,117	22,029,411	20,964,705	19,799,999	18,635,293	17,470,587
80	24,750,000	23,512,500	22,275,000	20,037,500	19,800,000	18,562,500
75	26,400,000	25,080,000	23,760,000	22,440,000	21,120,000	19,8000,00
70	28,285,714	26,871,427	25,457,142	24,042,856	22,628,571	21,214,285
65	30,461,538	28,938,461	27,415,384	25,892,307	24,369,230	22,846,153
60	33,000,000	31,350,000	29,700,000	28,050,000	26,400,000	24,750,000
58	34,137,931	32,431,034	30,724,137	29,017,241	27,310,344	25,603,448
56	35,357,142	33,589,284	31,821,427	30,053,570	28,285,713	26,517,856
54	36,666,666	34,833,332	32,999,999	31,055,556	29,222,222	27,499,999
52	38,076,923	36,173,076	34,269,230	32,365,384	30,416,538	28,557,692
50	39,600,000	37,620,000	35,640,000	33,660,000	31,680,000	29,700,000
49	40,408,163	38,387,754	36,367,346	34,346,938	32,326,530	30,306,122
48	41,250,000	39,187,500	37,125,000	35,062,500	33,000,000	30,937,500
47	42,127,659	40,021,276	37,914,893	35,808,510	33,702,127	31,595,744
46	43,043,478	40,891,304	38,759,130	36,586,956	34,434,782	32,282,608
45	44,000,000	41,800,000	39,600,000	37,400,000	35,200,000	33,000,000
44	45,000,000	42,750,000	40,500,000	38,250,000	36,000,000	33,750,000
43	46,046,511	43,744,285	41,441,959	39,139,534	36,837,208	34,534,883
42	47,142,857	44,785,714	42,428,571	40,071,428	37,714,285	35,357,142
41	48,292,682	45,878,047	43,463,413	41,048,779	38,634,145	36,219,511
40	49,500,000	47,025,000	44,550,000	42,075,000	39,600,000	37,125,000
39	50,769,230	48,230,768	45,692,307	43,153,845	40,615,384	38,076,922
38	52,105,263	49,499,999	46,894,736	44,289,473	41,684,210	39,978,947
37	53,513,513	50,837,837	48,162,161	45,486,486	42,810,810	40,135,134
36	55,000,000	52,250,000	49,500,000	46,750,000	44,000,000	41,250,000
35	56,571,642	53,743,059	50,914,477	48,085,895	45,257,313	42,428,731
34	58,235,294	55,323,529	52,411,764	49,499,999	46,588,235	43,676,470
33	60,000,000	57,000,000	54,000,000	51,000,000	48,000,000	45,000,000
32	61,875,000	58,781,250	55,687,500	52,593,750	49,500,000	46,406,250
31	63,870,967	60,677,418	57,483,870	54,290,321	51,096,773	47,903,225
30	66,000,000	62,700,000	59,400,000	56,100,000	52,800,000	49,500,000
29	68,275,862	64,861,968	61,448,275	58,034,482	54,620,689	51,206,896
28	70,714,285	67,178,570	63,642,850	60,107,142	56,157,428	53,035,713
27	73,333,333	69,666,666	65,999,999	62,333,333	58,666,666	54,999,999
26	76,153,846	72,346,153	68,538,481	64 730,769	60,923,076	57,115,384
25	79,200,000	75,240,000	71,280,000	67,320,000	63,360,000	59,400,000
24	82,500,000	78,375,000	74,250,000	70,125,000	66,000,000	61,875,000
23	86,086,956	81,782,608	77,478,260	73,173,912	68,869,564	64,565,217
22	90,000,000	85,500,000	81,000,000	76,500,000	72,000,000	67,500,000
21	94,285,714	89,571,428	84,857,142	80,142,856	75,428,571	70,714,285
20	99,000,000	94,050,000	89,100,000	84,150,000	79,200,000	74,250,000
19	104,210,526	98,999,999	93,789,473	88,578,947	83,368,420	78,157,894
18	110,000,000	104,500,000	99,000,000	93,500,000	88,000,000	82,500,000
17	116,470,588	110,647,048	104,823,529	98,999,999	93,176,470	87,352,941
16	123,750,000	117,582,500	111,375,000	105,187,500	99,000,000	92,812,500
15	132,000,000	125,400,000	118,800,000	112,200,000	105,600,000	99,000,000
14	141,428,571	134,357,142	127,285,713	120,214,285	113,142,856	106,071,428
13	152,307,692	144,692,307	137,076,922	129,461,538	121,846,153	114,230,769
12	165,000,000	156,750,000	148,500,000	140,250,000	132,000,000	123,750,000
11	180,000,000	171,000,000	162,000,000	153,000,000	144,000,000	135,000,000
10	198,000,000	188,100,000	178,200,000	168,300,000	158,400,000	148,500,000

Table Showing the

Comparative Steam Economy of Pumps

TYPE	Weight of Steam per 1 H. P. per hour
Fly-Wheel Triple Expansion condensing	13 to 16 lbs.
Fly-Wheel Compound High Speed with positive moved water valve	15 to 19 lbs.
Fly-Wheel Cross Compound—regular	18 to 20 lbs.
Single direct-acting Triple Compound condensing—large	23 to 24 lbs.
Single direct-acting Triple Compound condensing—small	25 to 27 lbs.
Burnham Single direct-acting Compound condensing—large	30 to 33 lbs.
Burnham Single direct-acting Compound condensing—small	35 to 38 lbs.
Duplex direct-acting Triple Compound condensing—large	26 to 28 lbs.
Duplex direct-acting Triple Compound condensing—small	28 to 30 lbs.
Duplex direct-acting Compound condensing—large	30 to 39 lbs.
Duplex direct-acting Compound condensing—small	40 to 43 lbs.
Duplex fly-wheel Simple condensing—large	40 to 43 lbs.
Duplex fly-wheel Simple condensing—small	45 to 48 lbs.
Duplex fly-wheel Simple non-condensing—large	48 to 50 lbs.
Duplex fly-wheel Simple non-condensing—small	52 to 55 lbs.
Burnham Single direct-acting Compound non-condensing —large	35 to 45 lbs.
Burnham Single direct-acting Compound non-condensing —small	45 to 55 lbs.
Duplex direct-acting Compound non-condensing—large	55 to 65 lbs.
Duplex direct-acting Compound non-condensing—small	65 to 75 lbs.
Burnham Single direct-acting ordinary—large	65 to 80 lbs.
Burnham Single direct-acting ordinary—small	80 to 100 lbs.
Duplex direct-acting ordinary—large	120 to 150 lbs.
Duplex direct-acting ordinary—small	150 to 200 lbs.

Capacity of Pumps at 100 Feet Piston Speed

*Theoretical Capacity of Pumps at 100 Feet Speed of Piston or Plunger

Diameter of Pump or Plunger in Inches	U. S. Gallons per			Diameter of Pump or Plunger in Inches	U. S. Gallons per		
	Minute	Hour	24 Hours		Minute	Hour	24 Hours
1	4.07	244.7	5875	14½	828	49704	1192896
1¼	6.37	382.5	9180	14½	858	51648	1235232
1½	9.18	550.8	13219	14¾	887	53256	1278144
1¾	12.49	749	17992	15	918	55070	1321915
2	16.31	979	23500	15¼	949	56928	1366272
2¼	20.6	1239	28180	15½	980	58800	1411200
2½	25.5	1530	36720	15¾	1012	60720	1457280
2¾	30.8	1851	44424	16	1044	62668	1504046
3	36.7	2203	52878	16¼	1077	64638	1551312
3¼	43.1	2586	62064	16½	1110	66642	1599408
3½	49.9	2998	71971	16¾	1144	68676	1648224
3¾	57.3	3442	82619	17	1179	70752	1698048
4	65.2	3916	94002	17¼	1214	72840	1748160
4¼	73.7	4422	106128	17½	1249	74964	1799136
4½	82.6	4957	118971	17¾	1285	77124	1850976
4¾	92	5523	132552	18	1322	79314	1903550
5	102	6120	146880	18¼	1359	81528	1956672
5¼	112	6745	161934	18½	1396	83778	2010672
5½	123	7404	177696	18¾	1434	86060	2065449
5¾	134	8093	194248	19	1473	88368	2120832
6	146	8812	211511	19¼	1511	90660	2175840
6¼	159	9562	229500	19½	1552	93120	2234880
6½	172	10344	248256	19¾	1590	95400	2289600
6¾	185	11152	267660	20	1632	97920	2350080
7	200	11995	287884	20¼	1673	100380	2409120
7¼	214	12867	308808	20½	1714	102840	2468160
7½	229	13769	330478	20¾	1756	105396	2529504
7¾	245	14700	352300	21	1799	107952	2590848
8	261	15667	376011	21¼	1842	110538	2652912
8¼	277	16660	399852	21½	1886	113154	2715696
8½	294	17688	424512	21¾	1930	115800	2779200
8¾	312	18741	449978	22	1974	118482	2843568
9	330	19828	475887	22¼	2020	121194	2908656
9¼	349	20944	502668	22½	2065	123924	2974176
9½	368	22092	530208	22¾	2111	126696	3040704
9¾	388	23280	558720	23	2158	129492	3107808
10	408	24480	587518	23¼	2205	132324	3175776
10¼	428	25716	617184	23½	2253	135186	3244464
10½	449	26989	647789	23¾	2301	138078	3313872
10¾	471	28290	678960	24	2349	140958	3382992
11	493	29616	710784	24¼	2399	143952	3454848
11¼	516	30986	743677	24½	2449	146958	3526992
11½	539	32374	776993	24¾	2499	149952	3598848
11¾	564	33795	811080	25	2550	152994	3671856
12	587	35251	846046	25½	2653	159179	3820300
12¼	612	36735	881640	26	2758	165484	3971630
12½	637	38250	918000	26½	2865	171908	4125800
12¾	663	39816	955584	27	2974	178457	4282967
13	689	41370	992880	27½	3085	185130	4443125
13¼	716	42972	1031328	28	3199	191922	4606125
13½	743	44610	1070640	28½	3314	198838	4772118
13¾	771	46278	1110672	29	3431	205876	4941028
14	799	47980	1151536	30	3672	220320	5287675

* For duplex pumps, the capacity given will be doubled.

Capacity of Cylinders

The following table is arranged to show the diameters, length of strokes in inches, with capacities per stroke in U. S. gallons, of Single-Acting Pump Cylinders. To ascertain the capacity of cylinders with longer stroke than listed, add or multiply the capacities of such lengths as will aggregate the length required. To ascertain the amount of water discharged per minute, multiply the capacity as rated per stroke by the number of strokes per minute.

Diam. Pump Cylin. In.	Length of Stroke in Inches, with Capacity of Cylinders, per Stroke, in U. S. Gallons																			
	1½	2	3	4	5	6	7	8	9	10	12	13	16	18	20	22	24	26	28	30
1	.0051	.0068	.0102	.0136	.0170	.0204	.0238	.0272	.0306	.0340	.0408	.0442	.0544	.0612	.0680	.0748	.0816	.0884	.0952	.1020
1¼	.0064	.0086	.0129	.0172	.0215	.0258	.0301	.0344	.0387	.0430	.0516	.0559	.0688	.0774	.0860	.0946	.1032	.1118	.1204	.1290
1½	.0079	.0106	.0159	.0212	.0265	.0318	.0371	.0424	.0477	.0530	.0636	.0689	.0848	.0954	.1060	.1116	.1272	.1378	.1484	.1590
1¾	.0096	.0128	.0192	.0256	.0320	.0384	.0448	.0512	.0576	.0640	.0768	.0832	.1024	.1152	.1280	.1408	.1536	.1664	.1792	.1920
1¾	.0114	.0152	.0228	.0304	.0380	.0456	.0532	.0608	.0684	.0760	.0912	.0988	.1216	.1368	.1520	.1672	.1824	.1976	.2128	.2280
1¾	.0133	.0178	.0267	.0356	.0445	.0534	.0623	.0712	.0801	.0 90	.1068	.1157	.1424	.1602	.1780	.1958	.2136	.2314	.2492	.2670
1¾	.0156	.0208	.0312	.0416	.0520	.0624	0728	.0832	.0936	.1041	.1248	.1452	.1664	.1872	.2082	.2288	.2496	.2904	.2912	.3123
1¾	.0178	.0238	.0357	.0476	.0595	.0714	.0833	.0952	.1071	.1190	.1428	.1547	.1904	.2142	.2380	.2618	.2856	.3094	.3332	.3570
2	.0204	.0272	.0408	.0544	.0680	.0816	.0952	.1088	.1224	.1360	.1632	.1768	.2176	.2448	.2720	.2992	.3264	.3536	.3808	.4080
2¼	.0258	.0344	.0516	.0688	.0860	.1032	.1204	.1376	.1548	.1720	.2064	.2236	.2752	.3096	.3440	.3784	.4128	.4472	.4816	.5160
2¼	.0318	.0424	.0636	.0848	.1060	.1272	.1484	.1696	.1908	.2120	.2544	.2756	.3392	.3816	.4240	.4664	.5088	.5512	.5936	.6360
2¾	.0385	.0514	.0771	.1028	.1285	.1542	.1799	.2056	.2313	.2570	.3084	.3341	.4112	.4626	.5140	.5654	.6168	.6682	.7196	.7710
3	.0459	.0612	.0918	.1224	.1530	.1836	.2142	.2448	.2754	.3060	.3672	.3978	.4896	.5508	.6120	.6732	.7344	.7956	.8568	.9180
3¼	.0538	.0718	.1077	.1436	.1795	.2154	.2513	.2872	.3231	.3590	.4308	.4667	.5744	.6462	.7180	.7898	.8616	.9334	1.0052	1.0770
3¼	.0624	.0832	.1248	.1664	.2080	.2496	.2912	.3328	.3744	.4160	.4992	.5408	.6656	.7488	.8320	.9152	.9984	1.0816	1.1648	1.2480
3¾	.0715	.0954	.1431	.1908	.2385	.2862	.3339	.3816	.4293	.4770	.5624	.6201	.7632	.8586	.9540	1.0494	1.1248	1.2402	1.3356	1.4310
4	.0814	.1086	.1629	.2172	.2715	.3258	.3801	.4344	.4887	.5430	.6516	.7059	.8688	.9774	1.0860	1.1946	1.3032	1.4118	1.5204	1.6290
4¼	.0919	.1226	.1839	.2452	.3065	.3678	.4291	.4904	.5517	.6130	.7356	.7969	.9808	1.1034	1.2260	1.3486	1.4712	1.5938	1.7164	1.8390
4¼	.1032	.1376	.2064	.2752	.3440	.4128	.4816	.5504	.6192	.6880	.8256	.8944	1.1008	1.2384	1.3760	1.5136	1.6512	1.7988	1.9264	2.0640
4¾	.1150	.1534	.2301	.3068	.3835	.4602	.5309	.6136	.6903	.7670	.9204	.9971	1.2272	1.3806	1.5340	1.6874	1.8408	1.9942	2.1476	2.3010
5	.1273	.1698	.2547	.3396	.4245	.5094	.5943	.6792	.7641	.8490	1.0188	1.1037	1.3584	1.5282	1.6980	1.8678	2.0376	2.2074	2.3772	2.5470
5¼	.1404	.1872	.2808	.3744	.4680	.5616	.6552	.7488	.8424	.9360	1.1232	1.2168	1.4976	1.6848	1.8720	2.0592	2.2464	2.4336	2.6208	2.8080
5¼	.1542	.2056	.3084	.4112	.5140	.6168	.7196	.8224	.9252	1.0280	1.2336	1.3364	1.6448	1.8504	2.0560	2.2616	2.4672	2.6728	2.8784	3.0840
6	.1834	.2446	.3669	.4892	.6115	.7338	.8561	.9784	1.1007	1.2230	1.4676	1.5899	1.9568	2.2014	2.4460	2.6906	2.9352	3.1798	3.4244	3.6690
7	.2497	.3330	.4995	.6660	.8325	.9990	1.1655	1.3320	1.4985	1.6650	1.9980	2.1645	2.6640	2.9970	3.3300	3.6630	3.9960	4.3290	4.6620	4.9950
8	.3262	.4350	.6525	.8700	1.0875	1.3050	1.5225	1.7400	1.9575	2.1750	2.6100	2.8275	3.4800	3.9150	4.3500	4.7850	5.2200	5.6550	6.0900	6.5250
9	.4129	.5506	.8259	1.1012	1.3765	1.6518	1.9271	2.2024	2.4777	2.7530	3.3036	3.5789	4.4048	4.9554	5.5060	6.0566	6.6072	7.1578	7.7084	8.2590
10	.5100	.6800	1.0200	1.3600	1.7000	2.0400	2.3800	2.7200	3.0600	3.4000	4.0800	4.4200	5.4400	6.1200	6.8000	7.4800	8.1600	8.8400	9.5200	10.2000
11	.6169	.8226	1.2339	1.6452	2.0565	2.4678	2.8791	3.2904	3.7017	4.1130	4.9356	5.3469	6.5808	7.4034	8.2260	9.0486	9.8712	10.6938	11.5164	12.3390
12	.7338	.9784	1.4676	1.9568	2.4460	2.9352	3.4244	3.9136	4.4028	4.8920	5.8704	6.3596	7.8272	8.8056	9.7840	10.7624	11.7408	12.7192	13.6976	14.6760
13	.8616	1.1488	1.7232	2.2976	2.8720	3.4464	4.0208	4.5952	5.1696	5.7440	6.8928	7.4672	9.1904	10.3392	11.4880	12.6368	13.7856	14.9344	16.0832	17.2320
14	.9993	1.3324	1.9986	2.6648	3.3310	3.9972	4.6634	5.3296	5.9958	6.6620	7.9944	8.6606	10.6592	11.9916	13.3240	14.6564	15.9888	17.3212	18.6536	19.9860
15	1.1475	1.5300	2.2950	3.0600	3.8250	4.5900	5.3550	6.1200	6.8850	7.6500	9.1800	9.9450	12.2400	13.7700	15.3000	16.8300	18.3600	19.8900	21.4200	22.9500
16	1.3050	1.7400	2.6100	3.4800	4.3500	5.2200	6.0900	6.9600	7.8300	8.7000	10.4400	11.3100	13.9200	15.6600	17.4000	19.1400	20.8800	22.6200	24.3600	26.1000
18	1.6518	2.2024	3.3036	4.4048	5.5060	6.6072	7.7084	8.8096	9.9108	11.0120	13.2144	14.3156	17.6192	19.8216	22.0240	24.2264	26.4288	28.6312	30.8336	33.0360
20	2.0400	2.7200	4.0800	5.4400	6.8000	8.1600	9.5200	10.8800	12.2400	13.6000	16.3200	17.6800	21.7600	24.4800	27.2000	29.9200	32.6400	35.3600	38.0800	40.8000
22	2.4681	3.2908	4.9362	6.5816	8.2270	9.8724	11.5178	13.1632	14.8086	16.4540	19.7448	21.3902	26.3264	29.6172	32.9080	36.1988	39.4896	42.7804	46.0712	49.3620
24	2.9370	3.9160	5.8740	7.8320	9.7900	11.7480	13.7060	15.6640	17.6220	19.5800	23.4960	25.4540	31.3280	35.2440	39.1600	43.0760	46.9920	50.9080	54.8240	58.7400
26	3.4473	4.5964	6.8946	9.1928	11.4910	13.7892	16.0874	18.3856	20.6838	22.9820	27.5784	29.8766	36.7712	41.3676	45.9640	50.5604	55.1568	59.7532	64.3496	68.9460
28	3.9979	5.3306	7.9959	10.6612	13.3265	15.9918	18.6571	21.3224	23.9877	26.6530	31.9836	34.6489	42.6448	47.9754	53.3060	58.6366	63.9672	69.2978	74.6284	79.9590
30	4.5895	6.1194	9.1791	12.2388	15.2985	18.3582	21.4179	24.4776	27.5373	30.5970	36.7164	39.7761	48.9552	55.0746	61.1940	67.3134	73.4328	79.5522	85.6716	91.7910

Ratios of Areas for Given Diameters of Steam and Water Pistons

Diameter of Water Piston or Plunger	DIAMETER OF STEAM CYLINDERS												
	3	4	4½	5	5½	6	6⅛	7	7⅛	8	8½	10	12
⅝	23.09	40.93	51.82	64.00	77.41	92.16	96.02	125.82	148.82	163.84	184.94	256.00	368.43
¾	16.00	28.44	35.99	44.44	53.76	64.00	66.69	87.11	103.37	113.77	128.45	177.77	256.00
⅞	11.75	20.90	26.44	32.65	39.49	47.02	48.99	63.99	75.93	83.59	94.36	130.61	188.10
1	9.00	16.00	20.24	25.00	30.23	36.00	37.50	48.99	58.13	64.00	72.24	100.00	144.00
1⅛	7.11	12.64	15.99	19.75	23.88	28.44	29.63	38.71	45.93	50.56	57.08	79.01	113.78
1¼	5.76	10.24	12.95	16.00	19.35	23.04	24.00	31.36	37.21	40.96	46.24	64.00	92.17
1⅜	4.76	8.47	10.70	13.22	15.99	19.04	19.83	25.91	30.74	33.85	38.20	52.89	76.17
1½	4.00	7.11	8.99	11.11	13.44	16.00	16.67	21.77	25.84	28.44	32.11	44.44	64.00
1⅝	3.40	6.06	7.66	9.46	11.45	13.63	14.20	18.55	22.01	24.23	27.35	37.87	54.54
1¾	2.93	5.22	6.61	8.16	9.87	11.75	12.24	16.00	18.09	20.99	23.59	32.65	47.02
1⅞	2.56	4.55	5.75	7.11	8.60	10.24	10.67	13.93	16.53	18.20	20.55	28.44	40.96
2	2.25	4.00	5.06	6.25	7.56	9.00	9.37	12.25	14.53	16 00	18.06	25.00	36.00
2¼	1.77	3.15	3.99	4.93	5.97	7.11	7.40	9.67	11.48	12.64	14.27	19.75	28.44
2½	1.44	2.56	3.23	4.00	4.83	5.76	6.00	7.84	9.30	10.24	11.56	16.00	23.04
2¾	1.19	2.11	2.69	3.30	3 99	4.76	4.96	6.47	7.68	8.46	9.55	13.22	19.04
3	1.00	1.77	2.24	2.77	3.36	4.00	4.16	5.44	6.46	7.11	8.02	11.11	16.00
3¼	.85	1.51	1.91	2.37	2.86	3.40	3.55	4.63	5.50	6.06	6.84	9.46	13.63
3½	.73	1.30	1.65	2.04	2.46	2.93	3.06	4.00	4.74	5.22	5.89	8.16	11.75
3¾	.64	1.13	1.44	1.77	2.15	2.56	2.66	3.48	4.13	4.55	5.13	7.11	10.24
4	.56	1.00	1.26	1.56	1.89	2.25	2.34	3.06	3.63	4.00	4.51	6.25	9.00
4¼	.49	.88	1.12	1.38	1.67	1.99	2.07	2.71	3.22	3.54	4.00	5.53	7.97
4½	.44	.79	1.00	1.23	1.49	1.77	1.85	2.42	2.87	3.15	3.35	4.93	7.11
4¾	.39	.70	.89	1.10	1.34	1.59	1.66	2.17	2.57	2.83	3.20	4.43	6.38
5	.36	.64	.80	1.00	1.20	1.44	1.50	1.96	2.32	2.56	2.89	4.00	5.76
5½	.29	.52	.66	.82	1.00	1.19	1.24	1.62	1.92	2.11	2.38	3.30	4 76
6	.25	.44	.56	.69	.84	1.00	1.04	1.36	1.61	1.77	2.00	2.77	4.00
6½	.....	.37	.47	.59	.71	.85	.88	1.15	1.37	1.51	1.71	2.37	3.40
7	.....	.33	.41	.51	.61	.73	.76	1.00	1.18	1.30	1.47	2.04	2.93
7½	.....	.28	.35	.44	.53	.64	.66	.87	1.03	1.13	1.28	1.77	2.56
8	.....	.25	.31	.39	.47	.56	.58	.76	.90	1.00	1.12	1.56	2.25
8½	.....	.....	.28	.34	.41	.49	.51	.67	.80	.88	1.00	1.38	1.99
9	.....	.....	.24	.30	.37	.44	.46	.60	.71	.79	.89	1.23	1.77
9½	.....	.....	.....	.27	.33	.39	.41	.54	.64	.70	.80	1.11	1.59
10	.....	.....	.....	.....	.30	.36	.37	.48	.58	.64	.72	1.00	1.44
10½	.....	.....	.....	.....	.27	.32	.34	.44	.52	.58	.65	.90	1.30
11	.....	.....	.....	.....	.....	.29	.31	.40	.48	.52	.59	.82	1.19
12	.....	.....	.....	.....	.....	.25	.26	.34	.40	.46	.50	.69	1.00
13	.....	.....	.....	.....	.....	.....	.....	.28	.34	.37	.42	.59	.85
14	.....	.....	.....	.....	.....	.....	.....	.25	.29	.35	.36	.51	.73
15	.....	.....	.....	.....	.....	.....	.....	.....	.25	.28	.32	.44	.64
16	.....	.....	.....	.....	.....	.....	.....	.....	.....	.25	.28	.39	.56
17	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.25	.34	.49
18	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.30	.44
20	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.25	.35
22	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.29
24	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.24
26	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
28	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
30	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
	3	4	4½	5	5½	6	6⅛	7	7⅛	8	8½	10	12

Ratios of Areas For Given Diameter of Steam and Water Pistons—Continued

Diameter Water Piston or Plunger	DIAMETER OF STEAM CYLINDERS											
	14	16	18	20	22	24	26	28	30	32	34	36
⅛	501.92	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
¼	348.51	455.11	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
⅜	256.00	334.37	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
1	196.00	256.00	324.00	400.00	.....	.....	.....	.....	.....	.....	.....	.....
1⅛	154.87	202.27	256.00	316.05	.....	.....	.....	.....	.....	.....	.....	.....
1¼	125.45	163.86	207.38	256.00	309.81	.....	.....	.....	.....	.....	.....	.....
1⅜	103.66	135.41	171.47	211.39	256.00	.....	.....	.....	.....	.....	.....	.....
1½	87.11	113.77	144.00	177.77	215.11	256.00	.....	.....	.....	.....	.....	.....
1⅝	74.24	96.96	122.72	151.54	183.37	218.22	.....	.....	.....	.....	.....	.....
1¾	64.00	83.59	105.79	130.61	158.05	188.10	220.71	.....	.....	.....	.....	.....
1⅞	55 75	72.82	92.16	113.78	137.67	163.85	192.29	.....	.....	.....	.....	.....
2	49.00	64.00	81.00	100.00	121.00	144.00	169.00	196.00	225.00	256.00	.....	.....
2⅛	38.71	50.56	64.00	79.01	95.60	113.78	131.56	154.87	177.77	202.27	.....	.....
2¼	31.36	40.96	51.84	64.00	77.44	92.16	108.01	125.44	144.00	163.84	184.97	.....
2⅜	25.91	33.85	42.84	52.89	64.00	76.17	89.39	103.66	119.01	135.41	152.86	.....
3	21.77	28.44	36.00	44.44	53.77	64.00	75.11	87.11	100.00	113.77	128.44	144.00
3⅛	18.56	24.23	30.67	37.87	45.83	54.54	64.00	74.24	85.22	96.96	109.46	122.72
3½	16.00	20.90	26.44	32.65	39.42	47.02	55.18	64.00	73.47	83.59	94.36	105.79
3⅜	13.93	18.20	23.04	28.44	34.42	40.96	48.07	55.75	64.00	72.82	82.21	92.16
4	12.25	16.00	20.25	25.00	30.25	36.00	42.25	49.00	56.25	64.00	72.25	81.00
4⅛	10.85	14.17	17.93	22.14	26.79	31.89	37.43	43.41	46.51	56.69	64.00	71.76
4¼	9.67	12.64	16.00	19.75	23.90	28.44	33.33	38.71	44.44	50.56	57.08	64.00
4⅜	8.68	11.34	14.36	17.73	21.45	25.53	29.96	34.75	39.89	45.38	51.24	57.44
5	7.84	10.24	12.96	16.00	1.19	23.04	27.04	31.36	36.00	40.96	46.24	51.84
5½	6.47	8.46	10.71	13.22	16.00	19.04	22.23	25.91	29.75	33.85	38.21	42.84
6	5.44	7.11	9.00	11.11	13.44	16.00	18.77	21.77	25.00	28.44	32.11	36.00
6½	4.63	6.06	7.66	9.46	11.45	13.63	16.00	18.56	21.30	24.23	27.36	30.67
7	4.00	5.22	6.61	8.16	9.87	11.75	13.79	16.00	18.37	20.90	23.59	26.44
7½	3.48	4.55	5.76	7.11	8.60	10.24	12.00	13.93	16.00	18.20	20.55	23.04
8	3.06	4.00	5.06	6.25	7.25	9.00	10.56	12.25	14.06	16.00	18.06	20.25
8½	2.71	3.54	4.48	5.53	6.69	7.97	9.35	10.85	12.45	14.17	16.00	17.92
9	2.41	3.15	4.00	4.93	5.85	7.11	8.34	9.67	11.11	12.64	14.27	16.00
9½	2.17	2.83	3.59	4.43	5.36	6.38	7.49	8.68	9.97	11.34	12.88	14.36
10	1.96	2.56	3.24	4.00	4.84	5.76	6.76	7.84	9.00	10.24	11.56	12.96
10½	1.77	2.32	2.93	3.62	4.38	5.22	6.13	7.11	8.16	9.26	10.48	11.75
11	1.62	2.11	2.67	3.30	4.00	4.76	5.58	6.47	7.43	8.46	9.55	10.71
12	1.36	1.77	2.25	2.77	3.36	4.00	4.67	5.44	6.25	7.11	8.02	9.00
13	1.16	1.51	1.91	2.37	2.86	3.40	4.00	4.63	5.32	6.06	6.83	7.66
14	1.00	1.30	1.65	2.04	2.46	2.93	3.44	4.00	4.59	5.22	5.89	6.61
15	.87	1.13	1.44	1.77	2.13	2.56	3.00	3.48	4.00	4.55	5.13	5.76
16	.76	1.00	1.26	1.56	1.89	2.25	2.64	3.06	3.51	4.00	4.51	5.06
17	.67	.88	1.12	1.38	1.67	1.99	2.34	2.71	3.11	3.54	4.00	4.48
18	.60	.79	1.00	1.23	1.49	1.77	2.08	2.41	2.77	3.15	3.56	4.00
20	.48	.63	.81	1.00	1.21	1.43	1.69	1.96	2.25	2.56	2.89	3.24
22	.40	.52	.66	.82	1.00	1.18	1.39	1.61	1.85	2.11	2.38	2.67
24	.34	.44	.56	.69	.84	1.00	1.17	1.36	1.56	1.77	2.00	2.25
26	.28	.37	.47	.59	.71	.85	1.00	1.15	1.33	1.51	1.71	1.91
28	.24	.32	.41	.51	.61	.73	.86	1.00	1.14	1.30	1.47	1.65
30	.....	.28	.35	44	.53	.63	.75	.87	1,00	1.13	1.28	1.44
	14	16	18	20	22	24	26	28	30	32	34	36

Valve Area

The valve area of a pump, which is generally expressed as a percentage of the water piston area, is the ratio of the total area of the effective suction or discharge valves on one stroke of the piston to the area of the water piston. The valve area is generally taken as the area through the valve seat on the assumption that the valve will lift a sufficient amount, so that the area measured at the periphery of the valve will be equivalent to the area of the opening in the valve seat. In a direct acting pump, there is generally the same number of suction valves and discharge valves, hence, valve area may refer to either suction or discharge area.

The valve area of pumps varies with the conditions of service; vacuum pumps, which handle a large per cent of air are generally given a valve area of 25 to 30 per cent. Boiler feed pumps generally have a valve area of 35 to 45 per cent. Pumps operating at 100 feet piston travel, and handling large volumes of water are generally given a valve area of 40 to 50 per cent. Elevator pumps, which usually operate at high speeds, have a valve area of 50 to 75 per cent.

Pump Valves

Direct acting pumps are generally fitted with rubber valves or bronze valves, depending on the service.

Medium rubber valves are used on pumps operating on low pressures up to 100 lbs. per square inch for handling cold water.

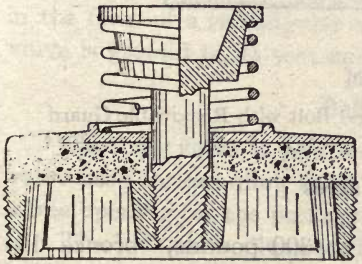


Fig. 124
Seat with Rubber Valve and one-piece Bolt.

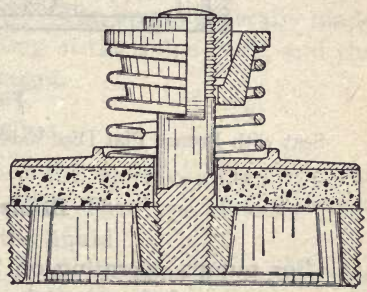


Fig. 125
Seat with Rubber Valve and Bolt with Removable Guard

Medium rubber valves are also used on wet vacuum pumps operating on a high vacuum. Hard rubber valves are used on pumps operating on 75 pounds to 200 pounds pressure per square inch, handling hot or cold water, and on vacuum heating pumps. Pumps handling very hot water 210° to 212° require special rubber valves, or bronze valves.

Figure 124 illustrates the rubber valve and its guard, seat, bolt and spring. The valve is guided by the bolt, which screws into the seat on a taper thread, and a bronze or iron guard is provided on top of the rubber valve in a bearing for the spring. The head of the valve bolt provides an upper guard for the spring. This head may be cast as part of the bolt, as shown in figure 124 or may be separate as shown in figure 126. The bolt with the removable guard is the type furnished for hand plate cylinders. The valve springs are bronze or steel, depending on how the pump is fitted. Rubber valves such as shown in figure 124 can easily be re-faced when worn.

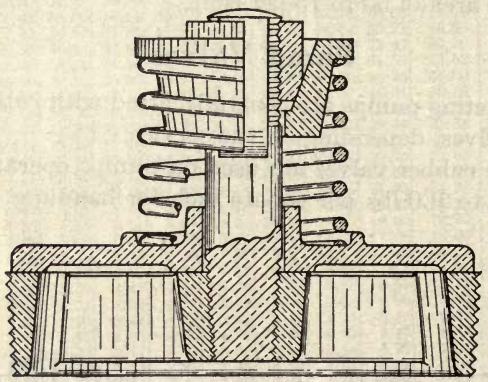


Fig. 126

Seat with Bronze Flat Disc Valve, and Bolt with Removable Guard

Bronze Valves

For pumps operating on 250 to 300 pounds, bronze discharge valves of the type shown in figure 126 are used. This type of valve can be easily inserted in place of the rubber valve if desired, and can be ground to a good seat.

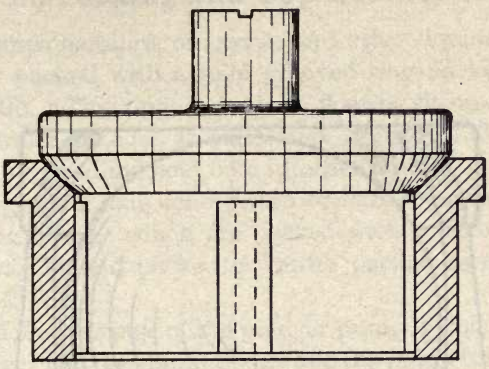
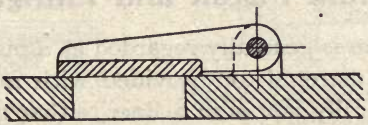


Fig. 127. Bevel Seat Wing Valve.

Beveled Seat Wing Valves

For high pressure pumps, the wing valve shown in figure 127 is used. This valve has a conical seat, and is provided with four wings, which guide the valve in its seat. This type of valve as a rule has a comparatively low lift, and can be easily ground to a seat under pressures up to 5000 pounds per square inch.

Fig. 128



Clapper Valve

Clapper Valves

For handling thick liquids, such as tar, molasses etc., valves with larger opening are necessary. For this purpose clapper valves may be used (see figure 128). The seat is generally made in the form of a rectangular opening in the valve deck, and the valve is ground to its seat and hinged.

Ball Valves

The ball valve is another and the more common type of valve used for handling thick liquids. This type of valve gives a free opening for the passage of the liquid.

The ball valve illustrated in figure 129 consists of a hollow ball made of either bronze or iron. The valve seat or cage may be screwed in place of a regular valve seat. The lift of the ball valve is limited by a cap, which screws into the top of the cage.

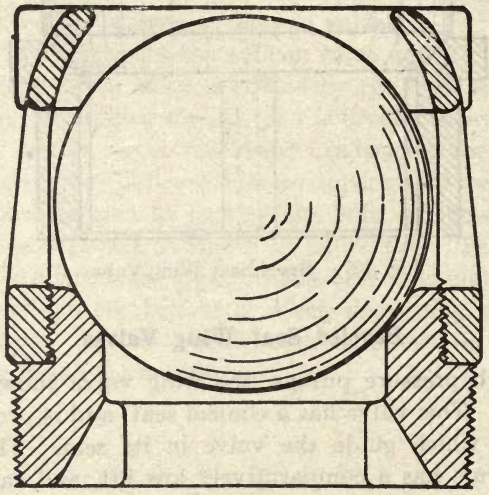


Fig. 129
Ball Valve and Cage.

Fluid Piston and Plunger

A fluid piston is generally regarded as a piston packed with either a fibrous material, or provided with metallic rings. A fluid piston generally works in a liner, either of bronze or iron depending upon the service.

The packed piston is extensively used, where the water or liquid to be pumped contains no grit or sand.

Figure 130a shows a piston packed with a fibrous material, such as is used on standard pumps.

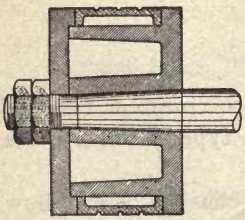


Fig. 130b.
Fluid Piston with
Ring-Grooved Packing

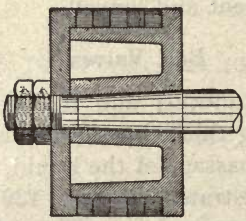


Fig. 130a.
Fluid Piston with
Fibrous Packing.

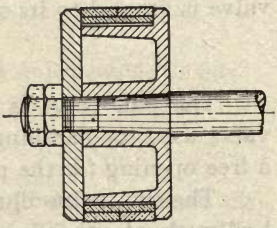


Fig. 204.
Fluid Piston with
Three-Ring Packing.

Metallic Packing and Cup Leather Pistons

For pumps handling oil, syrup and other liquids, the piston is generally packed with a solid grooved ring packing as shown in figure 130b. This ring is made a floating fit on the piston.

Cup leather pistons, as illustrated in figure 130c are sometimes used for handling cold oil and other liquids.

A plunger is a long solid piston or barrel working in one or two stuffing boxes, which are packed with a fibrous material. Plungers may be end packed or center packed as described on pages 271–272.

The chief advantage of the plunger pump is that any leakage by the plunger may be eliminated, while the pump is in operation.

The plunger pump is particularly suited for handling water or liquids containing sand and grit, and also for pressure pumps.

Stuffing Boxes

Small size pumps are generally furnished with screwed stuffing boxes as shown in figure 131a. The customary type of stuffing box used on pumps is shown in figure 131c which has a bolted gland.

Pumps operating on a high vacuum require a water-sealed

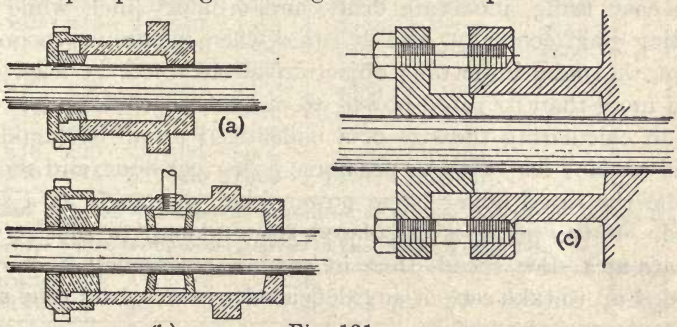


Fig. 131
Sketch Showing Plain Screwed Stuffing Box (a); Screwed Type with Lantern Gland (b); and Bolted Type (c).

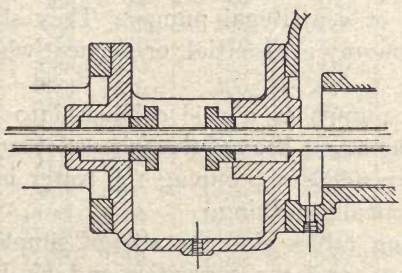


Fig. 132
Open Pot Water Seal Stuffing Box.

stuffing box on the water end. Two types are used for this purpose, the ordinary stuffing box with a lantern gland shown in figure 131 (b) and the open pot water seal shown in figure 132.

Pumps for Different Services

Boiler Feed Pumps

A Committee of the American Society of Mechanical Engineers recommended the unit of boiler power, known as the "Centennial Standard", and this is now generally accepted. They advised that the commercial horse power be taken as an evaporation of 30 pounds of water per hour from a feed water temperature of 100° of steam at 70 lbs. per square inch gauge pressure. This is practically equivalent to 34½ units of evaporation, that is, the 34½ lbs. of water evaporated from a feed-water temperature of 212° into steam at the same temperature. This "Centennial Standard" unit is equivalent to 33,317 British thermal units per hour.

It was the opinion of this committee that a boiler rated at any stated power should be capable of developing that power with easy firing, moderate draft, and ordinary fuel, while exhibiting good economy; and at times when maximum economy is not the most important object to be attained, at least one third more than its rated power, to meet emergencies.

In calculating the size of a boiler feed pump it should be based on 34½ lbs. of water per horse power per hour, and should handle the rated boiler horse power when operating at a slow speed. If the pump is calculated large enough so that it will operate at a slow speed, then in case of emergency it can be speeded up to take care of any deficiencies or overload that may be required of the boilers.

The types of pumps used for boiler feed service are either piston pumps, center packed plunger pumps, end packed plunger pumps, or centrifugal pumps. They should be compound where economy is essential, or vertical where floor space is limited.

Boiler feed pumps as a rule exhaust into a feed water heater. Thus the latent heat of the exhaust steam is recovered, and the heat expended in pumping the water into the boilers amounts to practically nothing.

The following tables give the size of simplex and duplex boiler feed pumps, and the capacities and H. P. boilers they are suitable for at various speeds.

Table of boiler horse-power capacities of Simplex Pumps at various speeds based on 34½ pounds per horse-power per hour. The maximum speed recommended for boiler-feed pumps is given in the table on page 292.

STROKES PER MINUTE	20	25	30	35	40	50	60
Size of Pump	Boiler Horse-Power Pumps Will Feed						
3 x 1¾x 3	....	.....	.....	.....	18	22	27
3 x 2 x 3	....	.....	.....	.....	23	29	35
4 x 2½x 3½	....	.....	32	38	43	54	65
4 x 2½x 5	....	38	46	54	62	77	92
5 x 3 x 6	53	66	80	93	106	132	160
5½x 3½x 7	85	106	126	147	168	210	253
6⅛x 4 x 8	126	158	190	220	253	315	378
7 x 4½x10	200	250	300	350	400	500	....
7 x 5 x10	245	308	370	430	490	615	....
8½x 5 x10	245	308	370	430	490	615	....
8½x 5½x10	295	370	445	520	595	740	....
10 x 6 x12	425	530	635	745	850	....	....
10 x 6½x12	500	685	750	875	1000	....	....
12 x 7 x12	580	725	870	1020	1160	....	....
12 x 7½x12	665	830	1000	1160	1330	....	....
12 x 8 x12	755	945	1135	1320	1500	....	....
14 x 8 x12	755	945	1135	1320	1500	....	....
14 x 8½x12	850	1070	1280	1500	1720	....	....
12 x 7 x16	770	970	1160	1350	1540	....	....
12 x 7½x16	885	1110	1330	1550	1780	....	....
12 x 8 x16	1010	1260	1510	1760	2020	....	....
14 x 8 x16	1010	1260	1510	1760	2020	....	....
14 x 8½x16	1140	1420	1710	2000	2280	....	....
14 x 9 x16	1275	1600	1920	2240	2550	....	....
16 x10 x16	1580	1980	2375	2760	3150	....	....
18 x12 x16	2275	2850	3400	4000	4500	....	....
20 x14 x16	3100	3875	4650	5400	6200	....	....
18 x12 x20	2840	3550	4250	5000	.....	....	....
20 x14 x20	3850	4820	5800	6750	.....	....	....
24 x16 x20	5050	6300	7500	8850	.....	....	....
26 x18 x20	6400	8000	9600	11200	.....	....	....
30 x20 x24	9500	11900	14200	16600	.....	....	....

Table of boiler horse-power capacities of Duplex Pumps at various speeds based on 34½ pounds per horse-power per hour. The maximum speed recommended for boiler-feed pumps is given in the table on page 292.

STROKES PER MINUTE*	20	25	30	35	40	50	60
Size of Pump	Boiler Horse-Power Pumps Will Feed						
2½x 1½x 3	....	.....	20	23	26	33	40
3 x 2 x 3	23	29	35	41	47	58	70
4½x 2¾x 4	59	74	89	100	118	148	178
5¼x 3½x 5	120	150	180	210	240	300	360
6 x 4 x 6	188	235	280	330	375	470	560
7½x 5 x 6	290	360	435	510	590	725	870
7½x 5 x 8	390	490	590	690	785	985	1180
7½x 4½x10	395	495	590	690	790	....	....
9 x 5¼x10	540	675	810	950	1080	....	....
10 x 6 x10	700	880	1060	1220	1400	....	....
10 x 6 x12	850	1060	1270	1490	1700	....	....
10 x 7 x10	960	1200	1450	1680	1925	....	....
12 x 7 x12	1160	1450	1740	2040	2320	....	....
12 x 8½x12	1700	2140	2560	3000	3440	....	....
14 x 8½x12	1700	2140	2560	3000	3440	....	....
14 x10 x12	2370	2950	3550	4150	4720	....	....
16 x10 x12	2370	2950	3550	4150	4720	....	....

*Each side.

Feed Pumps and Receivers

The receiver and pump is a desirable outfit for use in draining heaters, radiators, steam coils and steam jackets, and to force the water of condensation in its hottest condition direct to the boilers. This arrangement is entirely automatic in its action. Figure 133 illustrates the feed pump and receiver.

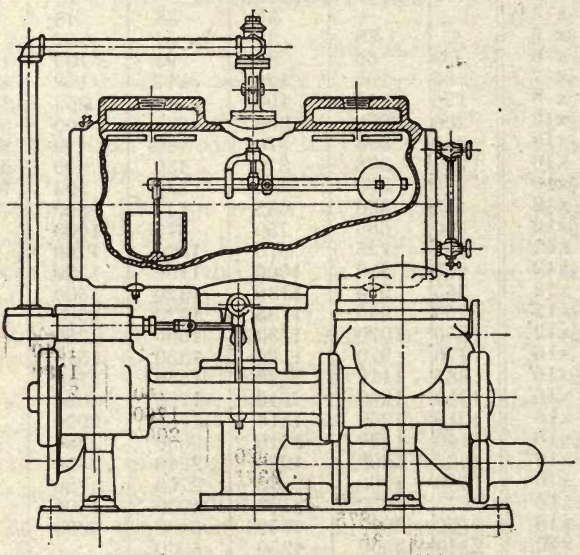


Fig. 133
Automatic Pump and Receiver.

The condensed steam enters at the top and flows by gravity into the receiver, which is provided with a float-control steam valve. As the float rises or falls, the speed of the pump is regulated, and the water is kept at a constant level in the receiver.

Should it be desired that this arrangement be the sole means of feeding the boilers, make-up water should be introduced directly into the receiver from which it is pumped into the boilers.

The size of the pump is calculated on the basis of one-half pound of steam condensed per hour per square foot of direct radiating surface.

Water Works Pumps

For water work service, direct acting pumps of either the piston or plunger pattern are used. As a rule a pump for this service is furnished with a compound steam end, as economy is generally of prime importance.

Water works pumps generally are controlled by a governor, which keeps the water pressure practically constant.

Mine Pumps

For mine service, two kinds of pumps are used, the sinking pump, and the station pump.

In sinking a mine, the first pump used is a sinking pump, illustrated in figure 134, which is of the center packed plunger type, and is so constructed that it can be lowered down in the mine shaft. It is secured to two steel hangers for hooking over a beam.

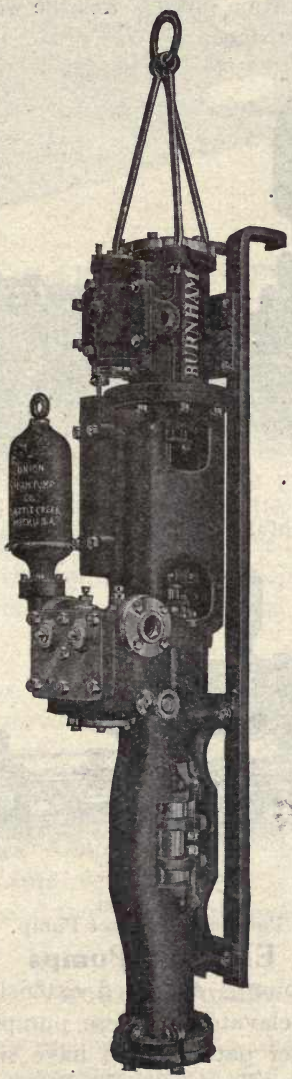


Fig. 134
Vertical Sinking Pump.

This pump is admirably adapted for this service, as it is very compact, and the valve gear and yoke are entirely enclosed, protecting it from falling rocks.

When a desired depth is reached in sinking the mine shaft, a chamber is cut out, and a station pump installed, which may be of the side plate piston pattern, as illustrated in figure 135, or of the pot valve plunger pattern, as illustrated in figure 109a.

A sump is provided into which the sinking pump delivers the water, and the station pump takes the water from the sump, and elevates to the surface.

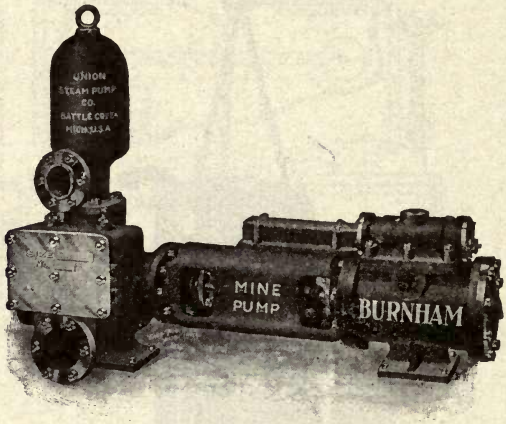


Fig. 135
Horizontal Mine Pump, Piston Pattern.

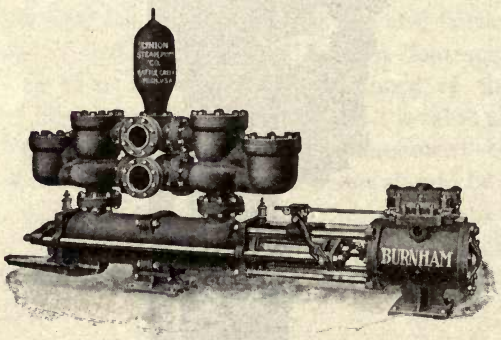


Fig. 109a
Pot Valve Plunger Pump.

Elevator Pumps

Direct acting pumps are used extensively for supplying water to hydraulic elevators. These pumps may be of either the piston or plunger pattern, and have simple or compound steam cylinders depending upon conditions. Figure 136 illustrates a compound center packed plunger elevator pump.

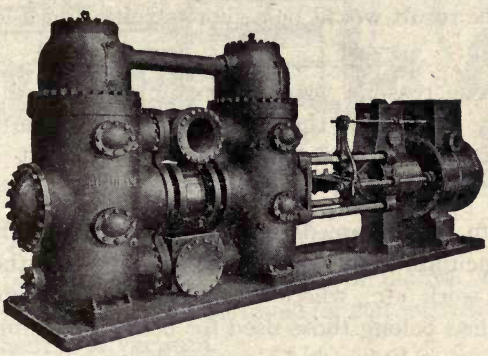


Fig. 136
Compound Center-Packed Plunger Elevator Pump.

Direct acting pumps are particularly suited for elevator service, as they are compact, occupy a small floor space, and are ready to start from any position as soon as steam is admitted to the cylinder. They merely require a pressure regulator or governor that keeps the pressure in the discharge tank uniform. Any sudden demand will lower the pressure in the discharge tank sufficiently to start the pump, and any stoppage of the elevator will increase the pressure in the discharge tank sufficiently to stop the pump.

Elevator pumps use the same water over and over again, and two tanks are provided for this purpose; a surge or suction tank, and a compression or discharge tank. The surge tank is generally located in the basement, and it may be closed or open. The discharge tank may also be of the closed or open type. If a closed or compression tank is used, the pump is controlled by a pressure regulator. The open tank is generally located in the roof, and the pump is controlled by a float.

Vacuum Pumps

For maintaining vacuums of 26″ and less, the direct acting vacuum pump in the simplex type is used extensively. Duplex pumps are not suitable for vacuum service, as they short stroke.

There being no resistance at the beginning of the stroke, except friction, one piston of a duplex pump would move rapidly forward, and throw the valve of the opposite pump before the piston of the opposite pump would have a chance to finish its stroke. The result would be a short stroke, and a low efficiency of the pump.

For these reasons the simplex pump is used for vacuum service. In this type, the piston cannot reverse until it has completed its stroke, and the result is an efficient and positive acting vacuum pump.

Vacuum pumps are divided into two classes, high vacuum and low vacuum. To the former class belong those used for condensing work etc., where a vacuum of 26″ is required, and to the latter class belong those used for heating systems etc., where a low vacuum of 10″ to 20″ is required.

High Vacuum Pumps

High vacuum pumps, which have small clearance, are provided with soft rubber valves, and have a water sealed stuffing box of either the lantern type or open pot type, as illustrated on page 313. This type of pump shown in figure 137a is used extensively in connection with jet and surface condensers, and vacuum pans, in removing the air and condensate.

The method of calculating the size vacuum pump to use in connection with jet and surface condensers has been clearly shown in Section Two on condensers.

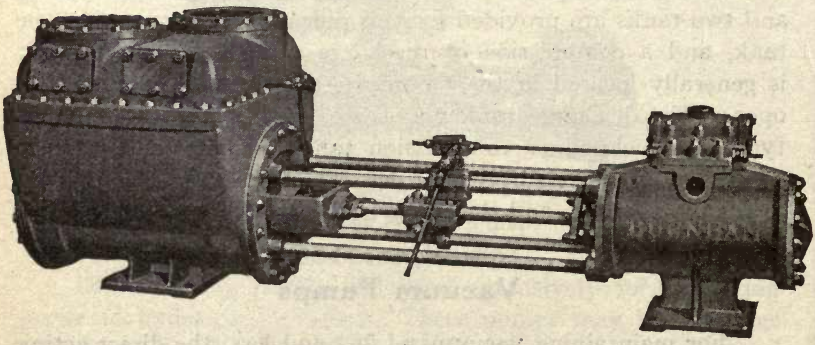


Fig. 137a
Inverted Suction Valve High Vacuum Pump.

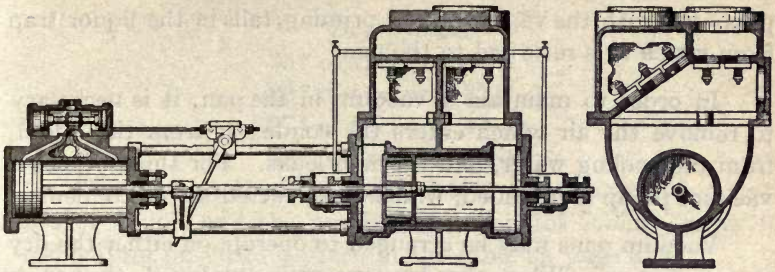


Fig. 137b
Section through Inverted Suction Valve, High Vacuum Pump.

Evaporation in a Vacuum

In sugar factories, as well as chemical plants and other industries, the modern method employed for concentrating liquors is by boiling them in a vacuum.

The advantages of evaporating in a vacuum over evaporating at atmospheric pressure are, first, that in a vacuum all liquids boil and evaporate at lower temperatures than under atmospheric pressure, thus there is a greater difference in temperature between the heating steam, and the boiling liquid, and consequently a much greater heat transmission.

Liquids that boil at high temperatures can generally not be evaporated under atmospheric pressure by means of high steam pressure, since steam would be required of such high temperatures and pressures, that its application would be dangerous. The boiling points of these liquids fall, when evaporated in a vacuum, so that steam of moderate pressures may be used.

The second advantage of boiling in a vacuum is that the liquid does not become as hot as at atmospheric pressure, and that also the heating surfaces, since steam of a lower pressure is used, are kept at a lower temperature. In most industries evaporating liquids, such as milk, gelatin, albumin etc., it is necessary in order not to discolor the liquids, that they be evaporated at low temperatures.

The ordinary form of vacuum pan comprises a spherical or cylindrical vessel, the lower portion of which is steam jacketed, and fitted with steam heating coils. At the upper portion of the vessel is a dome, which communicates through an exhaust pipe, provided with a liquor trap with a condenser, which in turn is connected with a vacuum pump. The steam and vapor given off from a charge of boiling liquor passes along the exhaust pipe to the condenser where it is condensed. Any liquor that might

pass over with the vapor, due to priming, falls in the liquor trap from which it is returned to the pan.

In order to maintain a vacuum in the pan, it is necessary to remove the air which enters the condenser from the liquid, from the cooling water, and through leaks. For this purpose a vacuum pump is provided, which is connected to the condenser.

Vacuum pans may be arranged to operate on either the dry or wet system. When the dry system is employed, the pan is fitted with a barometric condenser, and the vacuum pump handles only the noncondensable vapors. For this purpose a dry vacuum pump of the fly wheel type, as shown in figure 91, is generally used.

There are installations, however, operating on the dry system, which employ a wet vacuum pump, either of the fly wheel type, as shown in figure 176 or of the direct acting type, as shown in figure 137a. In cases of this kind the vacuum pump handles only the noncondensable vapors, and it is provided with a small quantity of charging water for sealing the valves. Figure 138 illustrates a vacuum pump with a barometric condenser operating on the dry system with a direct acting vacuum pump.

The vacuum in the pans operating on the dry system varies up to 28″, this factor depending upon the nature of the liquid to be evaporated.

Vacuum pans operating on the wet system are fitted with low level jet condensers, and the vacuum pump has to handle both the condensing water and the noncondensable vapors. For this service the vacuum pump is of the wet type, as shown in figure 137a, or of the fly wheel type as shown in figures 176—178. Vacuum pans operating on the wet system generally carry vacuum of 25″ to 26″.

The displacement of the vacuum pump for use with a vacuum pan on the wet or dry system is based on the amount of liquor to be evaporated.

The following figures give the approximate displacement of the vacuum pump based on the liquor to be evaporated.

Wet System

25″ of vacuum 60–1
26″ of vacuum 70–1

Dry System

25″ of vacuum 31–1
26″ of vacuum 37½–1
27″ of vacuum 47–1
28″ of vacuum 55–1

The amount of steam required to evaporate various quantities of liquor, and the number of gallons of cooling water for condensing purposes is given in the table on page 329.

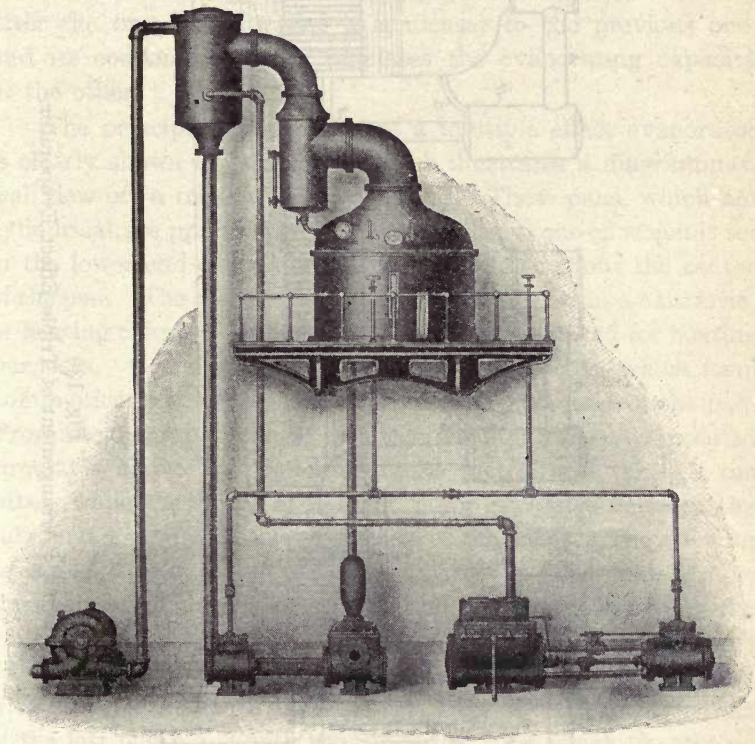


Fig. 138
Cut Showing a Vacuum Pan with Equipment of Pumps Operating on the Dry System with a Wet Vacuum Pump.

Multiple Effect Evaporator

Instead of evaporating a liquid in a single effect vacuum pan, quite often the process is distributed through several effects, two, three or four, with the result that there is a large saving made in the consumption of steam required to evaporate the liquor and consequently in the fuel necessary for the production of steam. This saving is more marked when the liquid to be concentrated is of a low density.

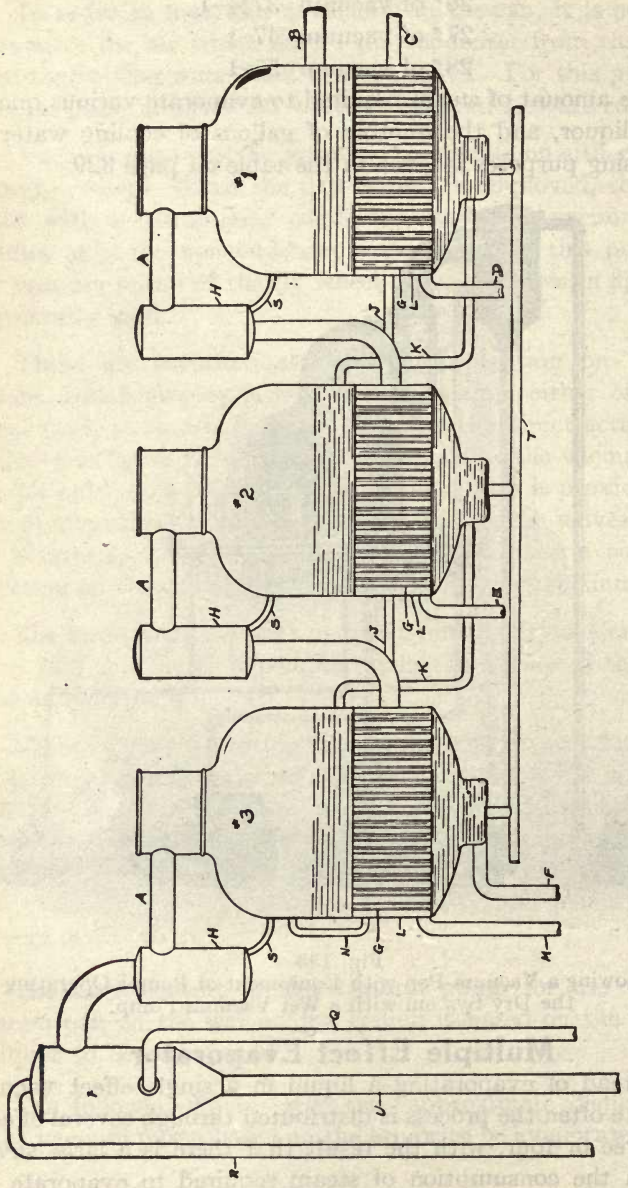


Fig. 139
Diagrammatic Arrangement of Triple Effect Evaporator.

The principle upon which the multiple effect pan works is the well known physical law that the latent heat of vapor is given off in condensing to a liquid, while the sensible heat is retained. Hence the employment of either live steam from the boiler or exhaust steam from the engine for heating the first pan or effect, and that resulting from the evaporation of the liquid itself that is introduced into the apparatus for concentration for heating the succeeding pans or effects, each effect after the first thus forming a condenser to the previous one, and its condensing power regulates the evaporating capacity of the other.

The principle of operation of a multiple effect evaporator is clearly shown in figure 139, which illustrates a diagrammatical view of a triple effect evaporator. These pans, which are cylindrical, are provided with two tube plates, one of which is set in the lower end of the pan, and the other at about the center of the pan. The space between the tube plates forms a calandria, or heating chamber L into which steam is introduced for heating purposes. G are tubes secured in the tube heads, which form communication between the upper or lower portions of the pan. From the diagram it can be seen that the liquid to be evaporated circulates above and below the tube plates, and through the tubes, while the steam or heating vapor circulates between the tube plates, and around the exterior of the tubes. The vacuum pans No. 1, No. 2 and No. 3 are partially filled with liquor, thus leaving in the upper part of each a space for receiving the vapor produced by the evaporation of the liquor. These spaces are connected by pipes A in the case of the pans No. 1 and No. 2 with the heating space L of the pan next in order, and in that of the last pan No. 3 with the condenser P, in which the vapors are condensed, and the air is drawn off by the air pump. The connecting pipes A between the pans are fitted with traps H to catch any of the liquor which might pass over with the vapor through priming, pipes S returning any liquor carried over from the traps to the pans. The liquor spaces of the calandria of the three pans are connected together by pipes K. C is the steam supply pipe to the heating space of the first pan No. 1, and B is the pipe for charging the first pan with liquor to be concentrated. T is a pipe which is connected by a suitable branch to a well in the bottom of each of the three effects, and by

means of which any one of the set can be emptied, if desired. M is a pipe for removing the concentrated syrup from the last effect No. 3. J is the by-pass pipe connecting the heating space of the calandria L of the No. 1, No. 2 and No. 3 pan. E. D. and M. are pipes for removing the condensed steam, or vapor from the calendria L. N is a by-pass pipe is the No. 3 pan connecting the upper part of the calendria L to the upper part of the pan, for removing the noncondensable vapors.

The operation of the multiple effect evaporator shown is as follows: The exhaust or waste steam is usually employed for heating the first effect No. 1. This steam is delivered into the heating space of the calandria L of part No. 1 where it circulates around the tubes G, and the condensation is drawn out through the pipe D, and is delivered to a tank from which it is returned to the boilers. The steam or vapor given off by the boiling liquor in pan No. 1 passes through pipe A to the heating space of the calandria L of the second pan No. 2, where it circulates around the tubes G, and then passes through the pipe J to the heating space L of the third pan No. 3, and together with the vapor given off by the boiling liquor in the second pan No. 2, which also passes to the heating space through the pipe A, circulates around the tubes G.

The noncondensable vapors are drawn off by the air pump from the heating space L of the No. 3 pan through the pipe N. The steam or vapor given off from the boiling liquor in the third or last pan No. 3 passes through the pipe A to the condenser P where it is condensed. The condensed steam or vapor and the condensing water fall through the tail pipe U, while the noncondensable vapors are withdrawn by the air pump through the pipe Q. The juice, or liquor passes through the apparatus from one pan to another through the pipes K, on account of the difference of vacuum, and the concentrated liquor is withdrawn from the last effect through the pipe F. by a thick liquor pump.

This apparatus is usually so arranged that it can be also worked as a double effect, or either pan singly if desired.

When employed as a triple effect evaporator, the first pan No. 1 generally has a vacuum of three or four inches, and a temperature of 200° ., the second pan No. 2 generally has a vacuum of about 14″, and a temperature of about 180°

and the third pan No. 3 generally has a vacuum of about 26″, and a temperature of about 125°.

It will be seen from this explanation that the effect is largely self heating, that is, the first pan only is heated by extraneous means, the latent heat of the vapor from the boiling liquor in the first pan being utilized to heat the liquor in the next, or second. The latent heat of the steam from the liquor in the second to heat that in the third. Not taking into consideration the loss of heat experienced by radiation, a double effect is twice, and the triplicate effect three times as economical of steam as the single effect.

Multiple effect evaporators are operated on both the wet and dry system, the same as vacuum pans, as described on page 322, and both the wet and dry vacuum pumps are employed, the type depending upon the installation.

While the displacement of the vacuum pump on a multiple effect evaporator depends on the number of effects, the figures given on page 322 may be used for calculating the displacement of the vacuum pump for multiple effect evaporators based on the total amount of liquor to be evaporated.

The amount of steam to use with multiple effect evaporators, as well as the quantity of condensing water is given in the table on page 329.

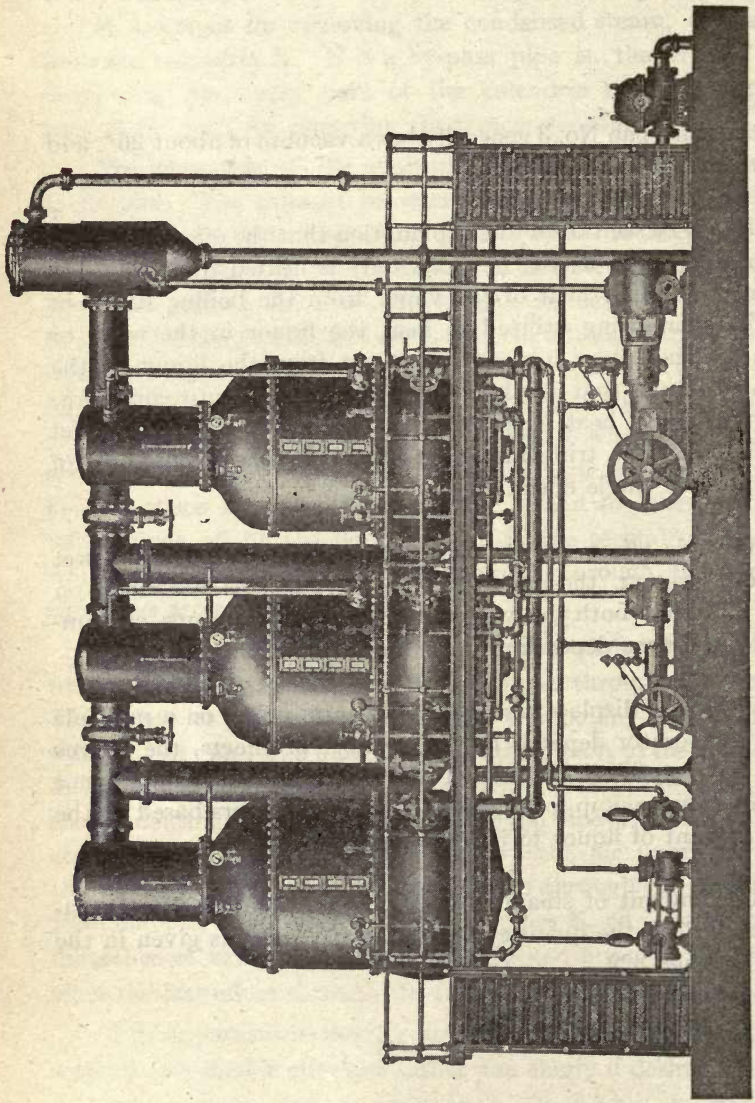


Fig. 140
Standard Triple Effect Evaporator Showing Equipment of Pumps.

Economy Rating of Evaporators*

Steam Consumption of Evaporators and Cooling Water Requirements of Condensers

**Total Evaporation in U. S. Gal. per Hour	‡Steam Consumed Pounds per Hour				†Cooling Water to Condenser—U.S.Gallons per Minute											
					Single Effect			Double Effect			Triple Effect			Quad. Effect		
	Single Effect	Double Effect	Triple Effect	Quadruple Effect	Temp. of Injection—°F.			Temp. of Injection—°F.			Tem. of Injection—°F.			Tem.of Injection—°F.		
					60	70	80	60	70	80	60	70	80	60	70	80
100	925	465	310	230	42	52	70	21	26	35	14	17	23	11	13	18
150	1390	695	465	345	63	78	105	32	39	53	21	26	35	16	20	26
200	1850	925	620	465	84	104	140	42	52	70	28	35	47	21	26	35
300	2775	1390	925	695	126	156	210	63	78	105	42	52	70	32	39	53
400	3700	1850	1230	925	168	208	280	84	104	140	56	69	95	42	52	70
500	4625	2315	1540	1160	210	260	350	105	130	175	70	87	115	53	65	88
750	6940	3475	2315	1730	290	360	480	160	195	265	105	130	175	79	98	130
1000	9250	4630	3080	2310	390	480	640	210	260	350	140	175	235	105	130	175
1250	11560	5780	3860	2890	490	600	800	245	300	400	175	215	290	130	165	220
1500	13880	6940	4630	3470	585	720	960	295	360	480	210	260	350	160	195	265
2000	18500	9250	6170	4630	780	960	1280	390	480	640	260	320	425	210	260	350
2500	23130	11570	7710	5790	975	1200	1600	490	600	800	325	400	535	245	300	400
3000	27750	13880	9250	6940	1170	1440	1920	585	720	960	390	480	640	295	360	480
4000	37000	18500	12340	9250	1560	1920	2560	780	960	1280	520	640	855	390	480	640
5000	46250	23130	15400	11560	1950	2400	3200	975	1200	1600	650	800	1065	490	600	800
6000	55500	27750	18500	13880	2340	2880	3840	1170	1440	1920	780	960	1280	585	720	960
7000	64750	32380	21580	16190	2730	3360	4480	1365	1680	2240	910	1120	1495	685	840	1120
8000	74000	37000	24670	18500	3120	3840	5120	1560	1920	2560	1040	1280	1705	780	960	1280
9000	83250	41630	27750	20810	3510	4320	5760	1505	2160	2880	1170	1440	1920	880	1080	1440
10000	92500	46250	30840	23130	3900	4800	6400	1950	2400	3200	1300	1600	2135	975	1200	1600
12000	111000	55500	37000	27750	4680	5760	7680	2340	2880	3840	1560	1920	2560	1170	1440	1920

Swenson Evaporator Co.

*Figures in this table *do not* represent the best performances of evaporators and condensers; but are compiled with reasonable allowance for the fluctuations and variations of average practice. Steam and cooling water requirements will often be found less and sometimes somewhat more, than indicated Steam consumption, for instance, varies with different efficiencies of heat insulating covering employed; while cooling water requirements are affected by air content of the injection water, and other factors. Any data obtained from this table will be found to agree closely with data generally obtained from actual operation.

**In the case of a multiple effect, figures given are for combined vaporation from all effects.

‡Amount of heating steam supplied to first effect, at pressure of 3 to 5 pounds, when liquor is fed to evaporator at approximately 120 degrees F. Different steam pressure, or reasonably different temperature of feed liquor will affect figures within a few percent only.

†Based on 26-inch vacuum (referred to 30-inch barometer) in last effect. Figures above heavy black line are for parallel current jet condensers, below heavy black line for counter current condensers. The type of condenser usually employed is as indicated by this statement.

*Steam Table for Evaporator Work

Vacuum—Referred to 30″ B Mercury at 58.4 Fahr.		Temperature		Specific Volume		Heat of the Liquid		Heat of Vaporization	
Inches	Millmeters	°Fahr.	°Cent.	Cu.Ft. per lb.	Cu.M. per Kilo	B.T.U. per lb.	Calories per Kilo	B.T.U pe lb..	Calories pr Kilo
29.0	736.6	79.07	26.11	657	41.02	47.11	26.17	1047.2	581.8
28.5	723.9	91.70	33.17	446.2	27.86	59.70	33.17	1040.3	577.9
28.0	711.2	101.15	38.42	339.6	21.20	69.12	38.40	1035.0	575.0
27.5	698.5	108.70	42.61	275.2	17.18	76.64	42.58	1030.8	572.7
27.0	685.8	115.06	46.14	231.9	14.48	82.98	46.10	1027.2	570.7
26.5	673.1	120.55	49.20	200.2	12.50	88.46	49.14	1024.1	568.9
26.0	660.4	125.38	51.88	176.7	11.03	93.28	51.82	1021.4	567.4
25.5	647.7	129.75	54.31	158.1	9.870	97.64	54.24	1018.9	566.1
25.0	635.0	133.77	56.54	143.0	8.927	101.65	56.47	1016.7	564.8
24.0	609.6	140.64	60.38	120.9	7.548	108.51	60.28	1012.8	562.7
23.0	584.2	146.78	63.77	104.5	6.524	114.64	63.69	1009.3	560.7
22.0	558.8	152.16	66.76	92.3	5.762	120.02	66.69	1006.2	559.0
21.0	533.4	157.00	69.44	82.6	5.157	124.86	69.37	1003.4	557.4
20.0	508.0	161.42	71.90	74.8	4.670	129.28	71.82	1000.8	556.0
19.0	482.6	165.42	74.12	68.5	4.276	133.28	74.04	998.5	554.7
18.0	457.2	169.14	76.19	63.1	3.939	137.00	76.11	996.4	553.6
17.0	431.8	172.63	78.13	58.6	3.658	140.50	78.06	994.3	552.5
16.0	406.4	175.93	79.94	54.6	3.409	143.80	79.89	992.3	551.3
15.0	381.0	179.03	81.69	51.2	3.196	146.91	81.62	990.5	550.3
14.0	355.6	181.92	83.29	49.0	3.059	149.80	83.22	988.8	549.3
13.0	330.2	184.68	84.82	45.55	2.844	152.57	84.76	987.1	548.4
12.0	304.8	187.31	86.28	43.2	2.699	155.21	86.23	985.5	547.5
11.0	279.4	189.83	87.68	41.05	2.563	157.73	87.63	984.0	546.7
10.0	254.0	192.23	89.02	39.1	2.441	160.14	88.97	982.2	545.6
9.0	228.6	194.52	90.29	37.4	2.335	162.44	90.24	981.2	545.1
8.0	203.2	196.73	91.52	35.8	2.235	164.68	91.49	979.8	544.3
7.0	177.8	198.87	92.70	34.3	2.141	166.81	92.67	978.6	543.7
6.0	152.4	200.94	93.86	33.0	2.060	168.88	93.82	977.2	542.9
5.0	127.0	202.92	94.96	31.8	1.985	170.89	94.93	976.1	542.3
4.0	101.6	204.85	96.03	30.6	1.910	172.81	96.06	974.8	541.6
3.0	76.2	206.71	97.06	29.55	1.845	174.68	97.04	973.7	540.9
2.0	50.8	208.52	98.07	28.6	1.785	176.50	98.06	972.6	540.3
1.0	25.4	210.28	99.04	27.7	1.729	178.27	99.04	971.4	539.7
0.0	0.0	212.00	100.00	26.8	1.673	180.00	100.00	970.4	539.1
Pressure									
Lbs. per sq. In. Gauge	Atmospheres								
1	1.068	215.3	101.8	25.23	1.575	183.4	101.9	968.2	537.9
2	1.136	218.5	103.6	23.80	1.486	186.6	103.7	966.2	536.8
3	1.204	221.5	105.3	22.53	1.407	189.6	105.3	964.3	535.7
4	1.272	224.4	106.9	21.40	1.336	192.5	106.9	962.4	534.7
5	1.340	227.2	108.4	20.38	1.272	195.3	108.5	960.6	533.7
6	1.408	229.8	109.9	19.45	1.214	198.0	110.0	958.8	532.7
7	1.476	232.4	111.3	18.61	1.162	200.6	111.4	957.2	531.8
8	1.544	234.8	112.7	17.85	1.114	203.1	112.8	955.2	530.7
9	1.612	237.1	113.9	17.14	1.070	205.4	114.1	954.0	530.0
10	1.680	239.4	115.2	16.49	1.029	207.7	115.4	952.5	529.2
15	2.021	249.7	120.9	13.88	.8660	218.2	121.2	945.5	525.3
20	2.361	258.8	126.0	11.99	.7485	227.4	126.3	939.3	521.8
30	3.041	274.1	134.5	9.45	.5900	243.0	135.0	928.5	515.8
40	3.722	286.7	141.5	7.82	.4882	255.9	142.2	919.4	510.8
50	4.402	297.7	147.6	6.68	.4169	267.2	148.4	911.2	506.2
60	5.082	307.3	152.9	5.83	.3640	277.1	153.9	903.9	502.2
70	5.763	316.0	157.8	5.18	.3234	286.1	158.9	897.2	498.4
80	6.443	323.9	162.2	4.67	.2915	294.3	163.5	891.0	495.0
90	7.124	331.2	166.2	4.24	.2647	301.8	167.7	885.3	491.8
100	7.804	337.9	169.9	3.89	.2429	308.8	171.6	880.0	488.9

*From tables given by Marks & Davis.

1 B. T. U. per pound=.5555 Calories per kilo. 1 Cal. per kilo=1.8 B. T. U. per pound.
1 Cu. Ft. per pound=.06243 Cu. Meters per kilo. 1 Cu Meter per Kilo=16.0184 Cu. Ft. per pound.

Boiling Points of Liquids (Degrees Cent.) at Atmospheric Pressure, and under various Degrees of Vacuum

LIQUID	Atmospheric Pressure	20.7 inches of Vacuum	24 inches of Vacuum	28 inches of Vacuum	29½ inches of Vacuum
Water	100	70	60.3	38.4	14.87
Alcohol	78.2	51	42	24	−3.1
Ether	35	5	−5	−25	−55
Acetic Acid	119.7	84.58	73.17	49.84	15
Benzine	80.36	46.6	35.36	12.86	−21
Oil of Turpentine	159	119.28	106	79.8	29.54
Butyric Acid	161.7	124.86	111.6	87	51.2
Glycerin	290	252.5	240	215	177.5
Mercury	357.25	297.25	277.25	237.25	177.25
Naphol	290	230	210	170	110
Carbolic Acid	178	142	130	104	70
Cresol	190	154	145	118	82

Useful Data

1 standard atmosphere (by definition) = 760 mm. of mercury at 0° Cent. = 29.921″ mercury at 32° F. = 30.000″ mercury at 58.4° F. = 33.90 ft. water at 32° F. = 14.696 pounds per sq. in.

Pressure Conversion Factors at 32° F.

	Inches Mercury	Feet Water	Pounds per sq. in.	Atmospheres
Inches Mercury	1	1.133	.491	.0334
Feet Water	.8827	1	.434	.0295
Lbs. per sq. in.	2.037	2.308	1	.0680

Approximate temperature correction factor for height of a mercury column = .0001 per degree F.

Example — Barometer reads 30.00″ at 80° F. To correct reading to 58.4° F. Correction = .0001 × 30 × (80—58.4) = 0 648″. Reading at 58.4° F. = 30.00″ + .065″ = 30.065″.

Reduction of Barometer to Sea Level

Mean Temperature of the Air Column 58°F.
Calculated from values given by Smithsonian Meteorological Tables

Elevation in feet	0	500	1000	2000	3000	4000	5000	6000	7000	8000	9000
Barometer reading in inches mercury at 58.4° F. equivalent to 30" reading at sea level.	30.	29.45	28.95	27.9	26.95	26.	25.1	24.2	23.3	22.5	21.7

Elevation in Feet at Various Points in the United States as Given by the U. S. Geological Survey

Albuquerque, N. M.	4943	Leadville, Colo.	10185
Atlanta, Ga	1032	Lexington, Ky.	946
Birmingham, Ala.	600	Minneapolis, Minn.	812
Boise, Ida.	2695	New York, N. Y.	54
Buffalo, N. Y.	583	Ogden, Utah	4296
Butte, Mont.	5576	Oklahoma City, Okla	1197
Cheyenne, Wyo	6062	Omaha, Neb.	1034
Chicago, Ill.	590	Phoenix, Ariz.	1082
Cincinnati, Ohio	490	Pittsburgh, Pa.	743
Dallas, Texas	425	Pocatello, Ida.	4461
Denver, Colo.	5183	Portland, Ore.	29
Des Moines, Ia.	803	Pueblo, Colo.	4660
Detroit, Mich.	579	Reno, Nev	4491
Duluth, Minn.	609	Salt Lake City, Utah.	4248
Helena, Mont.	4009	San Francisco, Calif.	15
Indianapolis, Ind.	708	Santa Fe, N. M.	6947
Kansas City, Mo.	750	Searles Lake, Calif.	1805
Knoxville, Tenn.	890	Spokane, Wash.	1909

Comparison of Various Hydrometer Scales

Degrees Baume	Specific Gravities: Standard adopted by U. S. Chem. Mfg. Ass. 15.5° Sp. gr. = 145.04 / (145.04 — B)	Specific Gravities: Modulus 144.38 Custom in France	Degrees Densimetric 15.5° C	Degrees Twaddell 60 Fahr. T°=200 (Sp. gr —1)	Degrees Brix. Official Prussian Hydrometer 15.6°C. Sp gr. = 400 / (400 — Bx. °	Degrees Beck 12.5°C. Sp. gr. = 170 / (170 — Bk. °	Degrees Brix Saccharimetric (Per cent, Sugar).	Gay-Lussac (Centigrade) Sp. gr. = 100 / (100 — C°
0	1.000	1.0000	0.0	0.0	0.0	0.0	0.0	0.0
1	1.007	1.0070	0.7	1.4	2.8	1.2	1.8	0.4
2	1.014	1.0140	1.4	2.8	5.5	2.3	3.6	1.4
3	1.021	1.0215	2.1	4.2	8.2	3.5	5.4	2.1
4	1.028	1.0285	2.8	5.6	10.9	4.6	7.1	2.7
5	1.036	1.0380	3.6	7.2	13.9	5.9	9.0	3.5
6	1.043	1.0435	4.3	8.6	16.5	7.0	10.7	4.1
7	1.051	1.0510	5.1	10.2	19.4	8.3	12.6	4.8
8	1.058	1.0585	5.8	11.6	21.9	9.3	14.3	5.5
9	1.066	1.0665	6.6	13.2	24.8	10.4	16.1	6.2
10	1.074	1.0745	7.4	14.8	27.5	11.7	18.0	6.9
11	1.082	1.0825	8.2	16.4	30.3	12.9	19.8	7.6
12	1.090	1.0905	9.0	18.0	33.0	14.1	21.5	8.3
13	1.098	1.0990	9.8	19.6	36.0	15.2	23.3	8.9
14	1.107	1.1075	10.7	21.4	39.0	16.4	25.2	9.7
15	1.115	1.1160	11.5	23.0	41.3	17.6	27.0	10.3
16	1.124	1.1245	12.4	24.8	44.2	18.8	28.9	11.0
17	1.133	1.1335	13.3	26.6	46.5	20.0	30.7	11.7
18	1.142	1.1425	14.2	28.4	49.7	21.2	32.6	12.4
19	1.151	1.1515	15.1	30.2	52.5	22.3	34.4	13.1
20	1.160	1.1607	16.0	32.0	55.2	23.5	36.2	13.8
21	1.169	1.1705	16.9	33.8	57.8	24.6	38.0	14.5
22	1.179	1.1795	17.9	35.8	60.7	25.8	40.0	15.2
23	1.188	1.1895	18.8	37.6	63.3	26.9	41.7	15.8
24	1.198	1.1995	19.8	39.6	66.1	28.1	43.6	16.5
25	1.208	1.2095	20.8	41.6	68.9	29.3	45.5	17.2
26	1.218	1.2195	21.8	43.6	71.6	30.4	47.3	17.9
27	1.229	1.2300	22.9	45.8	74.5	31.7	49.4	18.6
28	1.239	1.2405	23.9	47.8	77.2	32.8	51.2	19.3
29	1.250	1.2515	25.0	50.0	79.3	34.0	53.2	20.0
30	1.261	1.2625	26.1	52.2	82.8	35.2	55.1	20.7
31	1.272	1.2735	27.2	54.4	85.5	36.4	57.0	21.4
32	1.283	1.2850	28.3	56.6	88.3	37.5	58.9	22.1
33	1.295	1.2960	29.5	59.0	91.1	38.8	60.9	22.8
34	1.306	1.3080	30.6	61.2	93.7	39.9	62.7	23.4
35	1.318	1.3200	31.8	63.6	96.5	41.0	64.7	24.1
36	1.330	1.3320	33.0	66.0	99.2	42.2	66.7	24.8
37	1.342	1.3445	34.2	68.4	101.9	43.3	68.6	25.5

Comparison of Various Hydrometer Scales
(Continued)

Degrees Baume	Specific Gravities: Standard adopted by U. S. Chem. Mfg. Ass. 15.5° — Sp. gr. = 145.04 / (145.04 − B)	Specific Gravities: Modulus 144.38. Custom in France	Degrees Densimetric 15.5° C.	Degrees Twaddell 60 Fahr. T°=200 (Sp. gr.−1)	Degrees Brix. Official Prussian Hydrometer 15.6°C. Sp. gr. = 400 / (400 − Bx°)	Degrees Beck 12.5°C. Sp. gr. = 170 / (170 − Bk°)	Degree Brix Saccharimetric (Per cent Sugar).	Gay-Lussac (Centigrade) Sp gr. = 100 / (100 − C°)
38	1.355	1.3570	35.5	71.0	104.7	44.6	70.7	26.2
39	1.368	1.3700	36.8	73.6	107.6	45.8	72.7	26.9
40	1.381	1.3830	38 1	76.2	110.3	46.9	74.7	27.6
41	1.394	1.3955	39 4	78.8	113.5	48.0	76.7	28.3
42	1.408	1.4100	40.8	81.6	115.9	49.3	78.8	28.9
43	1.421	1.4240	42.1	84.2	118.5	50.4	80.8	29.6
44	1.436	1.4380	43.5	87.0	121.3	51.5	82.9	30.3
45	1.450	1.4525	45.0	90.0	124.1	52.8	85.1	31.0
46	1.465	1.4675	46.5	93.0	126.7	53.9	87.2	31.7
47	1.479	1.4827	48.0	96.0	129.7	55.1	89.4	32.4
48	1.495	1.4980	49.5	99.0	132.4	56.3	91.5	33.1
49	1.510	1.5135	51.0	102.0	135.1	57.4	93.6	33.8
50	1.526	1.5300	52.6	105.2	137.9	58.6	....	34.5
51	1.542	1.5460	54.2	108.4	140.6	59.8	....	35.2
52	1.559	1.5630	55.9	111.8	143.4	61.0	....	35.9
53	1.576	1.5800	57.6	115.2	146.2	62.2	....	36.6
54	1.593	1.5965	59.3	118.6	148.9	63.3	....	37.2
55	1.611	1.6150	61.1	122.2	151.7	64.5	....	37.9
56	1.629	1.6335	62.9	125.8	154.5	65.7	....	38.6
57	1.648	1.6520	64.8	129.6	157.3	66.9	....	39.3
58	1.666	1.6715	66.7	133.4	160.0	68.0	....	40.1
59	1.686	1.6910	68.6	137.2	162.8	69.2	....	40.7
60	1.706	1.7110	70.6	141.2	165.5	70.4	....	41.4
61	1.726	1.7315	72.6	145.2	168.3	71.5	....	42.1
62	1.747	1.7525	74.7	149.4	171.0	72 7	....	42.8
63	1.768	1.7740	76.8	153.6	173.8	73.8	....	43.4
64	1.790	1.7950	79.0	158.0	176.5	75.0	....	44.1
65	1.812	1.8185	81.2	162.4	179.3	76.2	....	44.8
66	1.835	1.8420	83.5	167.0	182 0	77.4	....	45.5
67	1.859	1.8660	85.9	171.8	184.8	78.6	....	46.2
68	1.883	1.8910	88.3	176.6	187.5	79.7	....	46.9
69	1.907	1.9151	90.7	181.4	190.2	80.9	....	47.6
70	1.933	1.9410	93.3	186.6	193.0	82.1	....	48.3
72.5	2.000	2.0085	100.0	200.0	200.0	85.0	....	50.0

Low Vacuum Pumps

Low vacuum pumps are mostly used on vacuum heating systems. This type of pump illustrated in figure 141a, when connected to the return pipe of a heating system, insures a positive circulation of steam throughout the system, and does away with the annoying cracking or hammering in the pipe line. This application not only increases the efficiency of the system, but greatly reduces the cost of operation, and returns the condensed steam at a high temperature for feeding back to the boilers.

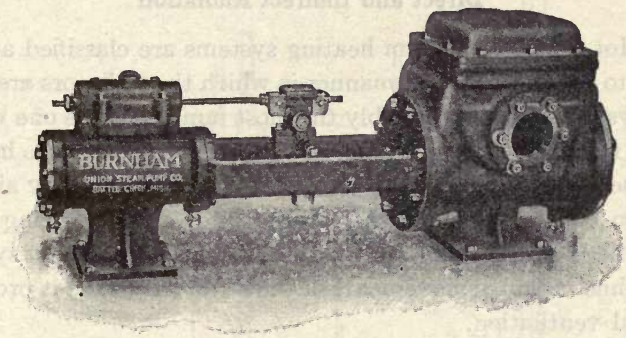


Fig. 141a
Low Vacuum Pump

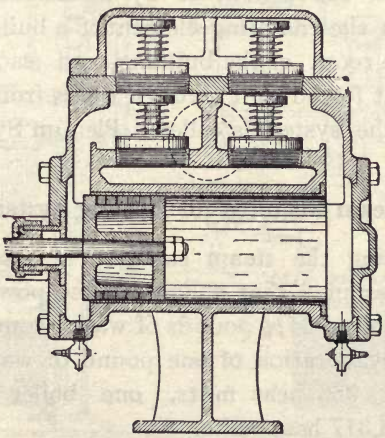


Fig. 141b
Section through Vacuum Cylinder of Low Vacuum Pump

The low vacuum pump, however, is not intended to deliver the water of condensation directly to the boilers, but should deliver to an open heater, or tank, and the water may then be returned directly to the boiler by means of a regular boiler feed pump. Where there is no heater, the vacuum pump may discharge into the receiver of an automatic pump and receiver as described on page 316. The condensate is then automatically returned to the boilers without further attention. Figure 142 illustrates a diagram of a vacuum heating system, and the vacuum pump and boiler feed pump connections are clearly shown.

Direct and Indirect Radiation

Hot water and steam heating systems are classified accordingly to the position and manner in which the radiators are used. The system which is probably the most familiar is the one wherein the radiators are placed directly within the space to be heated. This heating is accomplished by direct radiation, and by air convection currents through the radiator, no provision being made for a change of air in the room. This is known as a direct system, and while it causes movements of the air in the room, it produces no real ventilation.

In the indirect system, the heat radiating element is erected somewhat distant from the rooms to be heated, and ducts carry the heated air from the radiator to the rooms heated either by natural convection, or by means of fan or blower pressure. When the radiating element of a building is installed together in one room of the building, and each room has its share of the heat forced to it through ducts from one centralized fan or blower, the system is called a Plenum System.

Steam Required for Heating Systems

In calculating the steam capacity necessary for heating systems, it is assumed that a boiler horse power is equivalent to the evaporation of 34½ pounds of water from and at 212°.

Since the evaporation of one pound of water from and at 212° . requires 966 heat units, one boiler horse power is equivalent to 33,317 heat units.

For heating purposes, a more convenient standard of power is the number of square feet of radiating surface. Carpenter

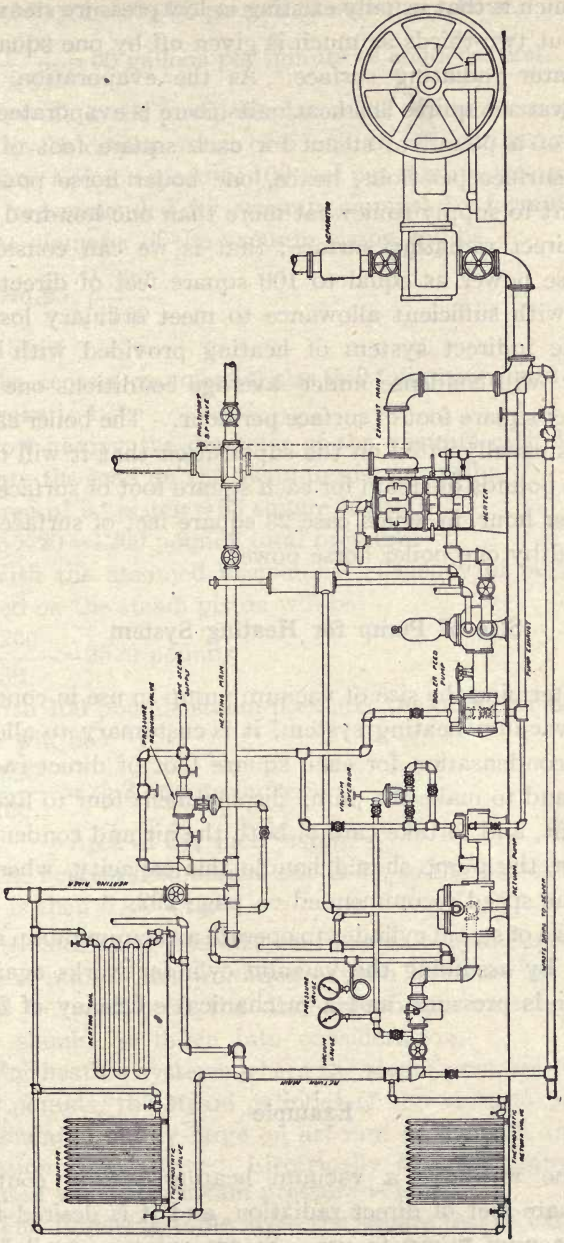


Fig. 142
Design of a Vacuum Heating System.

says that each square foot of direct radiating surface gives off 220-280 heat units per hour, when the difference of temperature is 150°, which is that usually existing in low pressure steam heating. About two-thirds as much is given off by one square foot of hot water radiating surface. As the evaporation of one pound of water requires 966 heat units, there is evaporated about one-third of a pound of steam for each square foot of steam radiating surface per hour, hence, one boiler horse power will be sufficient to supply somewhat more than one hundred square feet of direct radiating surface; that is we can consider the boiler horse power as equal to 100 square feet of direct steam radiation with sufficient allowance to meet ordinary losses.

In the indirect system of heating provided with blower, the heater will condense under average conditions one pound of water per square foot of surface per hour. The boiler capacity required is usually rated on the supposition that it will need to supply 1.5 pounds of steam for each square foot of surface in the radiator per hour, in which case 23 square feet of surface would be supplied by one boiler horse power.

Size of Pump for Heating System

To determine the size of vacuum pump to use in connection with the vacuum heating system, it is customary to allow one pound of condensation for each square foot of direct radiation per hour, and to make the pump displacement four to five times this amount, and to take care of both the air and condensate in the system, the pump should handle this capacity, when operating at the speed recommended on page 292.

The size of steam cylinder to operate a vacuum pump may be calculated by assuming the vacuum cylinder works against 15 to 20 pounds pressure and a mechanical efficiency of 50 per-cent.

Example

Assume we have a vacuum heating system containing 30,000 square feet of direct radiation, and it is desired to find the size vacuum pump to use. Steam pressure available 100 pounds.

Solution

$30{,}000 \times 1 = 30{,}000$ pounds of steam per hour.

$\frac{30{,}000}{8.3 \times 60} = 60$ gallons per minute of condensation.

Now allow the pump displacement of five to one, the pump must have a displacement of 5×60, or 300 gallons per minute. Assuming a piston speed of 100 feet per minute (see page 292 for speeds recommended for vacuum pumps) by formula 51, page 293, the diameter of the vacuum piston will be

$$d = 4.95\sqrt{\frac{300}{100}}$$

$= 8\frac{5}{8}''$

The nearest commercial size is 9″ diameter, and the standard stroke is 12″.

Now having the diameter of the vacuum cylinder, we may calculate the load on the vacuum piston thusly,

Area of 9″ piston = 63 square inches.

$63 \times 20 = 1260$ pounds total pressure.

With the assumed mechanical efficiency of 50%, the load required on the steam piston will be

$\frac{1260}{.50} = 2520$ pounds

and with 100 pounds steam pressure, the diameter of the steam piston will be

$\frac{2520}{100} = 25.2$ Square inches

= Area of $5\frac{3}{4}$ inch piston.

The nearest commercial size is 6″, so the size of the vacuum pump is then $6 \times 9 \times 12$.

In calculating the size of the steam cylinder to use on a vacuum pump, the minimum steam pressure to be carried should be used, and the discharge head, including frictional losses should be taken into consideration.

For heating systems where the steam pressure runs from 10 to 20 pounds, the steam cylinder of the vacuum pump should be calculated amply large on account of the fact that the condensation is excessive. Electrically driven Pumps are generally used where the steam pressure is low.

The following table gives the square feet of external radiation for different sizes of pipe.

Square Feet of Radiating Surface of Pipe per Lineal Foot

On all lengths over one foot, fractions less than tenths are added to or dropped.

Length of Pipe	Size of Pipe											
	¾	1	1¼	1½	2	2½	3	4	5	6	7	8
1	.275	.346	.434	.494	.622	.753	.916	1.175	1.455	1.739	1.996	2.257
2	.5	.7	.9	1.	1.2	1.5	1.8	2.4	2.9	3.5	4.	4.5
3	.8	1.	1.3	1.5	1.9	2.3	2.7	3.5	4.4	5.2	6.	6.8
4	1.1	1.4	1.7	2.	2.5	3.	3.6	4.7	5.8	7.	8.	9.
5	1.4	1.7	2.2	2.4	3.1	3.8	4.6	5.8	7.3	7.7	10.	11.3
6	1.6	2.1	2.6	2.9	3.7	4.5	5.5	7.	8.7	10.5	12.	13.5
7	1.9	2.4	3.	3.4	4.4	5.3	6.4	8.2	10.2	12.1	14.	15.8
8	2.2	2.8	3.5	3.9	5.	6.	7.3	9.4	11.6	13.9	16.	18.0
9	2.5	3.1	3.9	4.4	5.6	6.8	8.2	10.6	13.1	15.7	18.	20.3
10	2.7	3.5	4.3	4.9	6.2	7.5	9.1	11.8	14.6	17.4	20.	22.6
11	3.	3.8	4.8	5.4	6.8	8.3	10.	12.9	16.	19.1	22.	24.9
12	3.3	4.1	5.2	5.9	7.5	9.	11.	14.1	17.4	20.9	24.	27.1
13	3.6	4.5	5.6	6.4	8.1	9.8	11.9	15.3	18.9	22.6	26.	29.4
14	3.8	4.8	6.1	6.9	8.7	10.5	12.8	16.5	20.3	24.3	28.	31.6
15	4.1	5.2	6.5	7.4	9.3	11.3	13.7	17.6	21.8	26.1	30.	33.9
16	4.4	5.5	6.9	7.9	10.	12.0	14.6	18.8	23.2	27.8	32.	36.1
17	4.7	5.9	7.4	8.4	10.6	12.8	15.5	20.	24.7	29.5	34.	38.4
18	5.	6.2	7.8	8.9	11.2	13.5	16.5	21.2	26.2	31.3	36.	40.6
19	5.2	6.6	8.3	9.4	11.8	14.3	17.4	22.3	27.6	33.1	38.	42.9
20	5.5	6.9	8.7	9.9	12.5	15.	18.3	23.5	29.1	34.8	40.	45.2
21	5.8	7.3	9.1	10.4	13.	15.8	19.2	24.7	30.5	36.5	42.	47.4
22	6.	7.6	9.6	10.9	13.7	16.5	20.2	25.9	32.	38.3	44.	49.7
23	6.3	8.	10.	11.3	14.3	17.3	21.1	27.	33.5	40.	46.	52.
24	6.6	8.3	10.4	11.9	14.9	18.	22.	28.2	34.9	41.7	48.	54.2
25	6.9	8.6	10.9	12.3	15.6	18.8	22.9	29.3	36.3	43.5	50.	56.4
26	7.1	9.	11.3	12.8	16.2	19.5	23.8	30.5	37.8	45.2	52.	58.6
27	7.4	9.4	11.7	13.3	16.8	20.3	24.7	31.7	39.3	47.	54.	61.
28	7.7	9.7	12.2	13.8	17.4	21.	25.6	32.9	40.7	48.7	56.	63.2
29	8.	10.	12.6	14.3	18.	21.8	26.6	34.1	42.2	50.4	58.	65.5
30	8.3	10.4	13.	14.8	18.7	22.5	27.5	35.3	43.6	52.1	60.	67.7
31	8.5	10.7	13.5	15.3	19.3	23.3	28.4	36.4	45.1	53.9	62.	70.
32	8.8	11.1	13.9	15.8	19.9	24.1	29.3	37.6	46.5	55.6	64.	72.2
33	9.1	11.4	14.3	16.3	20.5	24.8	30.2	38.8	48.	57.4	66.	74.4
34	9.4	11.7	14.7	16.8	21.2	25.6	31.1	40.	49.5	59.1	68.	76.7
35	9.6	12.1	15.2	17.3	21.8	26.3	32.	41.1	50.9	60.8	70.	79.
36	9.9	12.5	15.6	17.8	22.4	27.	33.	42.3	52.4	62.6	72.	81.3
37	10.2	12.8	16.1	18.3	23.	27.8	33.9	43.5	53.8	64.3	74.	83.5
38	10.5	13.2	16.5	18.8	23.7	28.5	34.8	44.6	55.2	66.	76.	85.8
39	10.7	13.5	16.9	19.3	24.3	29.3	35.7	45.8	56.7	67.8	78.	88.
40	11.	13.8	17.4	19.8	24.9	30.1	36.6	47.	58.2	69.5	80.	90.2
41	11.3	14.2	17.8	20.3	25.5	30.8	37.6	48.2	59.6	71.3	82.	92.5
42	11.5	14.5	18.2	20.8	26.1	31.6	38.5	49.4	61.1	73.	84.	94.8
43	11.8	14.9	18.7	21.3	26.8	32.3	39.4	50.6	62.5	74.8	86.	97.
44	12.1	15.2	19.1	21.8	27.4	33.1	40.3	51 7	64.	76.5	88.	99.3
45	12.4	15.6	19.5	22.2	28.	33.8	41.2	52.9	65.5	78.2	90.	101.6
46	12.7	15.9	20.	22.7	28.6	34.6	42.2	54.	67.	80.	92.	103.8
47	12.9	16.3	20.4	23.2	29.2	35.3	43.	55.2	68.4	81.7	94.	106.
48	13.2	16.6	20.8	23.7	29.9	36.1	43.9	56.4	69.8	83.5	96.	108.4
49	13.5	17.	21.3	24.2	30.5	36.8	44.8	57.6	71.2	85.1	98.	110.5
50	13.8	17.3	21.7	24.7	31.1	37.6	45.8	58.7	72.7	87.	100.	112.8

Hydraulic Pressure Pumps

Hydraulic pressure pumps are generally used with hydraulic presses, and accumulators in connection with hydraulic presses.

For this work a pump of small capacity is generally required and the pressure against which it operates varies from 500 up to 5000 lbs. per square inch. For pressures up to 2000 lbs. per square inch the hydraulic pressure pump with cast iron cylinder is used, and for pressures above this a hydraulic pump with either a cast or forged steel cylinder is used.

The purpose of the accumulator is to furnish a momentary demand for water, which the hydraulic pressure pumps can replenish, when the hydraulic press is not in operation. The accumulator thereby provides an elastic element in the system, which will maintain a constant supply to the presses, acting as a cushion against shocks, and giving the pump an opportunity to get in motion before there is any marked drop in pressure.

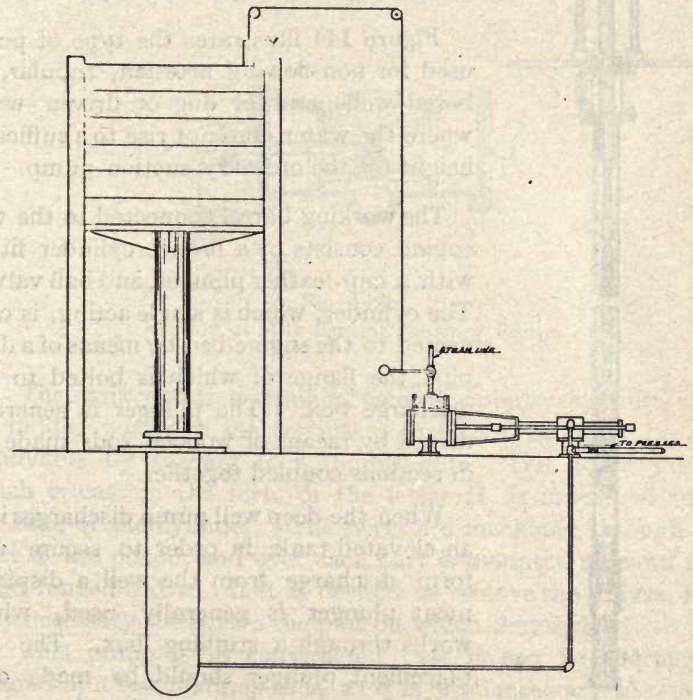


Fig. 143.
Sketch Showing Accumulator and Hydraulic Pressure Pump.

The accumulator consists of a vertical cylinder, the upper end of which is provided with a stuffing box through which a plunger or ram works. This ram carries a platen loaded with heavy weights, which are equivalent to the area of the ram or plunger multiplied by the water pressure.

Figure 143 illustrates an accumulator and a hydraulic pump. The governing device which operates the pump is so arranged that the pump will maintain the ram at its highest position. The steam line to the pump is fitted with a butterfly valve, which is operated by a balanced lever. When the ram reaches its highest position, it trips the weight W, which is connected to the chronometer valve, and stops the pump. When the demand for water increases, the ram descends, and with it the weight W, starting up the pump.

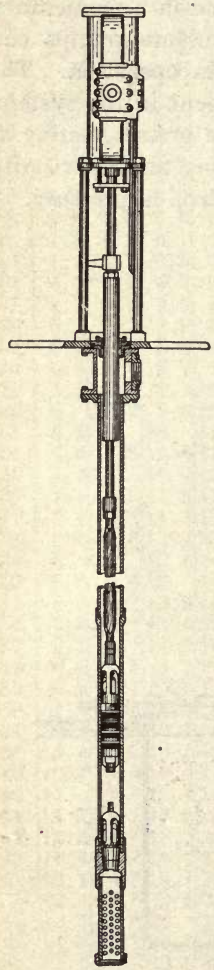


Fig. 144
Deep Well Pump.

Deep Well Pumps

Figure 144 illustrates the type of pump used for non-flowing artesian, tubular, or bored wells, and for dug or driven wells, where the water does not rise to a sufficient height for the ordinary suction pump.

The working barrel connected to the well engine consists of a bronze cylinder fitted with a cup leather plunger, and ball valves. The cylinder, which is single acting, is connected to the engine bed by means of a drop pipe, the flange of which is bolted to the discharge box. The plunger is generally driven by means of wooden rods made up in sections coupled together.

When the deep well pump discharges into an elevated tank, in order to secure uniform discharge from the well, a displacement plunger is generally used, which works through a stuffiing box. The displacement plunger should be made one-half the area of the well plunger to secure uniform discharge.

Milk Pumps

For handling milk and other liquid food products, it is imperative that a strictly sanitary type of pump should be used. Figure 145a illustrates a direct acting milk pump. Figure 145b shows the construction of the liquor end.

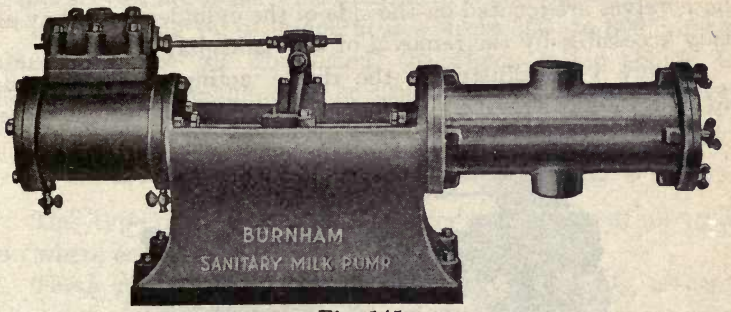


Fig. 145a
Sanitary Pump.

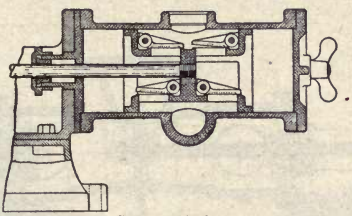


Fig. 145b
Section of Milk End of Sanitary Pump.

The milk end is a straight, smooth, tubular cylinder with no hidden parts, or inaccessible corners. The outer head is removable by turning four thumb nuts. The liquor piston, which is cast in the form of the letter H is machined to fit the bore of the cylinder. The ports are machined through the sides of the piston, and over each part is mounted a round flat-faced hinged valve. If it is desired to remove the valves, it is only necessary to remove the pins on which they swing.

This pump is so constructed that it can be thoroughly cleaned in a few minutes time, as it is only necessary to remove one bolt from the crosshead. The entire assembled piston and valves can then be removed, and immersed in water.

Magma Pumps

This type of pump is designed for handling thick and heavy liquids, such as massecuite, etc. The liquor cylinder is made without suction valves, and the material flows by gravity into the cylinder through a rectangular port.

The discharge takes place at both ends of the cylinder through large flat faced valves, having free opening seats. These valves are located on the side of the cylinder, and they are easily accessible by the removal of the hand plates.

Figure 146a illustrates the direct acting magma pump, and figure 146b shows the construction of the liquor cylinder.

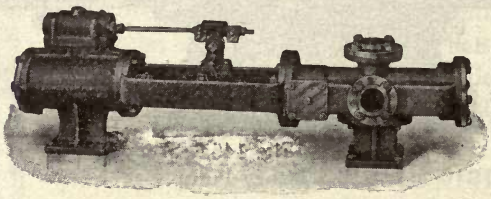


Fig. 146a
Magma Pump.

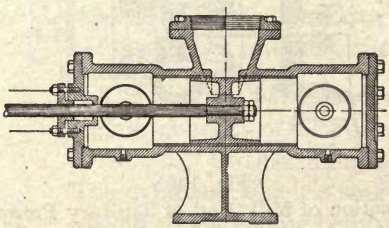


Fig. 146b
Section through Magma Cylinder..

Oil Pumps

Direct acting pumps are extensively used for handling oil.

For pipe line work, the pot valve plunger pump is generally used, as the friction head which this class of pump operates against is generally around 500 to 1000 lbs. per square inch.

In oil refineries, etc., for handling cold oil, the piston or plunger pump is used. For this service, the pump is generally brass fitted, and the piston packed pump is fitted with ring packing, or cup leathers, as illustrated on page 312.

Pumps for handling hot oil in refineries may be of the piston or plunger type. Where the temperature of the oil does not exceed 300° . a regular fitted pump is used. Piston pumps for this service are fitted with brass ring piston packing.

For handling oil of temperatures from 300 to 800° ., the pump should be all iron fitted, the piston pump should be fitted with ring packing, and the stuffing boxes should be packed with asbestos packing.

Direct Acting Air Compressors

This type of compressor is used for low pressures up to 40 lbs., where economy of operation is not essential.

These compressors are used for agitating liquids, creating vacuum etc. Figure 147 illustrates a direct acting air compressor.

For higher air pressures than 40 lbs., the fly type air compressor fully described in Section One is used.

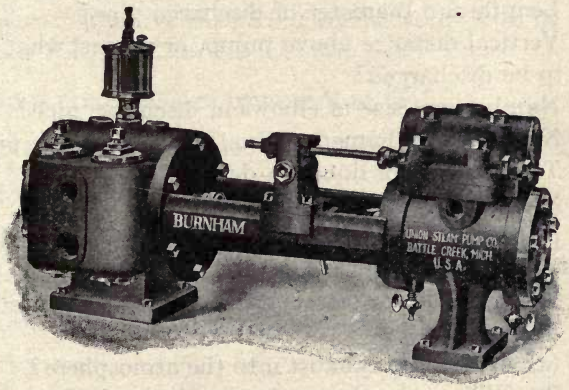


Fig. 147
Direct Acting Air Compressor.

Data Required for Estimates for Direct Acting Steam Pumps

When sending for estimates, please answer the following questions:

1. For what purpose is pump to be used?
2. (a) Capacity of pump in U. S. gallons per minute?

(b) If the pump is for vacuum service, give the number of square feet of radiation, or the number of cubic feet displacement per minute required?

(c) If the pump is for use with a condenser, give the number of pounds of steam per hour to be condensed, temperature of condensing water, the vacuum to be carried and the type of condenser?

(d) If pump is for evaporator, give the nature of the liquid to be evaporated, the quantity of liquor to be evaporated per hour, the temperature of the condensing water, the vacuum under which the liquid is to be evaporated, and the number of the effects in the evaporator?

3. Total lift, including suction, discharge, lift, and pipe friction in feet?
4. Length and diameter of the suction pipe?
5. Vertical distance from water level to pump in feet?
6. Number and size of elbows in suction pipe?
7. Length and diameter of discharge pipe?
8. Vertical distance above pump, or against what pressure is liquid to be discharged?
9. Number and size of elbows in discharge pipe?
10. Number and diameter of valves in discharge pipe?
11. Temperature of liquid in degrees Fah.?
12. Specific gravity of liquid?
13. Nature of liquid to be handled: fresh water, salt water, acidulous, alkaline, gritty, etc.?
14. What is the lowest steam pressure to be used at the pump?
15. (a) Will pump exhaust into the atmosphere?

(b) Will pump exhaust into a heater? (State whether open or closed).

16. What pressure will pump exhaust against?
17. If pump is to operate condensing, give the vacuum to be carried on the condenser?
18. Where is pump to be located, on the surface, or underground?

Data Required for Estimates for Deep Well Pumping Engines

When sending for estimates, please answer the following questions:

1. What is the entire depth of well? .....

2. What is the inside diameter of casing?.....

3. If boring is reduced, state at what depth and to what diameter? .....
cased with.....inside diameter casing to a depth of.....feet; balance of well cased with..... inside diameter casing.

4. Depth from surface to water level when not pumping .....

5. Capacity of well when pumped.....gallons per minute.

6. Depth from surface to water level when pumped at this capacity.....

7. Style and capacity of pump used.....

8. Elevation above surface to which water is to be raised .....

9. Horizontal distance from well to tank.....

10. Steam pressure carried at boiler.....

11. How far from well is boiler located?.....

12. What is the lowest steam pressure you want pump to operate with?.....

13. How many gallons of water do you require per hour? .....

14. Have you a water cylinder already in well?.....
and at what depth?.....

15. If so, what is the diameter?.....
Length of stroke?.....

Burnham Horizontal Piston Pattern Boiler Feed or Pressure Pumps

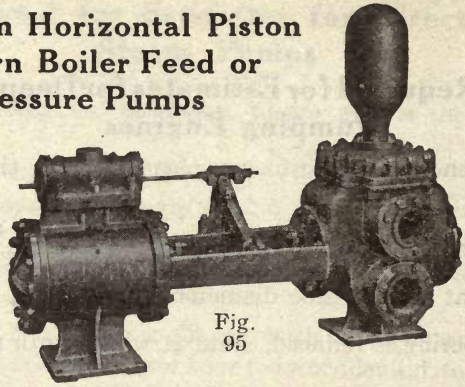


Fig. 95

In the sizes listed below, 14x8x12 and smaller are suitable for a Maximum Working Pressure of 250 Pounds of Steam and Water. Larger sizes are suitable for a Maximum Working Pressure of 150 Pounds Steam and Water.

Size of Pump			Diam. Pipe Openings				Ratings			
			For long pipe lines use larger pipes, reducing size at the pump openings							
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Gallons per Stroke	Max. Strokes per Minute	Capacity G.P.M., Maximum Speed	H.P. Boilers Pump will feed at Slow Speed
4	2½	5	½	¾	1½	1¼	.106	140	14.8	62
5	3	6	½	¾	2	1½	.183	130	23.8	106
5½	3½	7	½	¾	2½	2	.291	120	35	147
6⅛	4	8	¾	1	2½	2	.44	115	50.7	220
7	4½	10	¾	1	3	2½	.688	108	74.1	350
7	5	10	¾	1	3½	3	.85	108	92	430
8½	5	10	1	1½	3½	3	.85	108	92	430
8½	5½	10	1	1½	3½	3	1.02	108	111	520
10	6	12	1¼	2	4	3	1.46	100	146	635
10	6½	12	1¼	2	4	3	1.72	100	172	750
12	7	12	1½	2½	5	4	2.00	100	200	870
12	7½	12	1½	2½	5	4	2.29	100	229	1000
12	8	12	1½	2½	5	4	2.61	100	261	1135
14	8	12	2	2½	5	4	2.61	100	261	1135
14	8½	12	2	2½	6	5	2.94	100	294	1280
12	7	16	1½	2½	5	4	2.66	75	200	1160
12	7½	16	1½	2½	5	4	3.05	75	229	1330
12	8	16	1½	2½	5	4	3.48	75	261	1510
14	8	16	2	2½	5	4	3.48	75	261	1510
14	8½	16	2	2½	6	5	3.93	75	294	1710
14	9	16	2	2½	6	5	4.40	75	330	1920
16	10	16	2	2½	8	6	5.44	75	400	2375
18	12	16	2½	3½	8	6	7.82	75	587	3400
20	14	16	2½	3½	10	8	10.66	75	800	4650
18	12	20	2½	3½	8	6	9.78	60	587	4250
20	14	20	2½	3½	10	8	13.32	60	800	5800
24	16	20	3	5	10	8	17.40	60	1044	7500
26	18	20	3	5	12	10	22.00	60	1322	9600
30	20	24	3	5	14	12	32.64	50	1632	14200

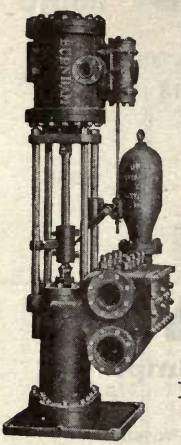


Fig. 112a

Burnham Vertical Piston Pattern Boiler Feed or Pressure Pumps

300 Pounds Maximum Steam and Water Pressure

Size of Pump			Diameter of Pipe Openings				Ratings			
			For long pipe lines use larger pipes, reducing size at the pump openings							
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Gallons per Stroke	Max Strokes per Minute	Capacity G. P. M., Max. Speed	H. P. Boilers Pump will feed at Slow Speed
4	2½	5	½	¾	1½	1¼	.106	140	14.8	62
5	3	6	½	¾	2½	2	.183	130	23.8	106
5½	3½	7	½	¾	2½	2	.291	120	35	147
6⅛	4	8	¾	1	3	2½	.44	115	50.7	220
7	4½	10	¾	1	3	2½	.688	108	74.1	350
8½	5	10	1	1½	3½	3	.85	108	92	430
8	5	12	1	1½	3½	3	1.01	100	101	440
10	6	12	1¼	2	4	3	1.46	100	146	635
10	6	16	1¼	2	4	3	1.95	75	146	840
10	7	12	1½	2	5	4	2.00	100	200	870
12	7	12	1½	2½	5	4	2.00	100	200	870
12	7	16	1½	2½	5	4	2.66	75	200	1160
12	7	18	1½	2½	5	4	2.99	66⅔	200	1300
12	8	12	1½	2½	5	4	2.61	100	261	1135
12	8	16	1½	2½	5	4	3.48	75	261	1510
12	8	18	1½	2½	5	4	3.92	66⅔	261	1700
12	8	24	1½	2½	5	4	5.22	50	291	2280
14	8	12	2	2½	5	4	2.61	100	261	1135
14	8	16	2	2½	5	4	3.48	75	261	1510
14	8	18	2	2½	5	4	3.92	66⅔	261	1700
14	8	24	2	2½	5	4	5.22	50	261	2280
14	9	16	2	2½	6	5	4.40	75	330	1920
14	9	18	2	2½	6	5	4.95	66⅔	330	2150
14	9	24	2	2½	6	5	6.60	50	330	2870
16	10	16	2	2½	6	5	5.44	75	400	2375
16	10	18	2	2½	6	5	6.12	66⅔	400	2670
16	10	24	2	2½	6	5	8.16	50	400	3550

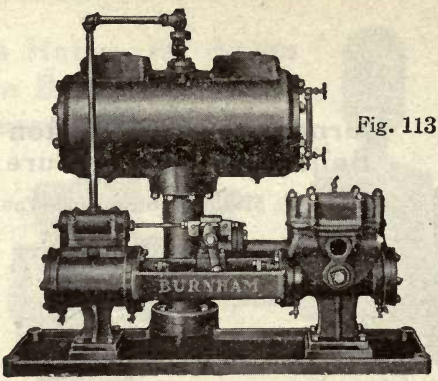


Fig. 113

Burnham Automatic Feed Pumps and Receivers

250 pounds Maximum Steam and Water Pressure
Receivers suitable for 150 Pounds Maximum Pressure

Size of Pump			Size of Openings					CAPACITY	
			For long pipe lines use larger pipes, reducing size at the pump openings						
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exaust	Suction	Discharge	Drain Inlets	Gallons per Minute at Speed Recommended	Square feet of Radiating Surface
4	2½	5	½	¾	1½	1¼	2–2″	8.5	8500
5	3	6	½	¾	2	1½	2–2″	13.5	13500
5½	3½	7	½	¾	2½	2	2–2″	20	20000
6⅛	4	8	¾	1	2½	2	2–3″	30	30000
7	4½	10	¾	1	3	2½	2–3″	41	41000
8½	5	10	1	1½	3½	3	2–3″	51	51000
10	6	12	1½	2	4	3	2–6″	73	73000
12	7	12	1½	2½	5	4	2–6″	100	100000
12	8	12	1½	2½	5	4	2–6″	130	130000

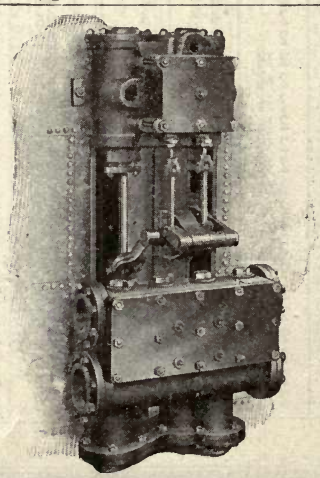


Fig. 114

Union Vertical Duplex Piston Pattern Boiler Feed or Pressure Pumps

200 pounds Maximum Steam and Water Pressures

SIZE	Openings				Ratings			
	Steam	Exhaust	Suction	Discharge	Gallons per Stroke Each Cylinder	Max. Stroke Each Cylinder per Minute	Capacity G. P. M., Both Cyls. Maximum Speed	H. P. Boilers Pump will feed at a Slow Speed
4½x2¾x4	½	¾	2	1½	.103	150	30	125
5¼x3½x5	¾	1¼	2½	1½	.208	140	58	300
6 x4 x6	1	1¼	3	2	.326	130	84	400
7½x5 x6	1½	2	4	3	.510	130	133	600
7½x4½x10	1½	2	4	3	.689	96	132	700
9 x5¼x10	2	2½	4	3	.938	96	180	850
10 x6 x10	2	2½	5	4	1.224	96	235	1200
10 x7 x10	2	2½	6	5	1.66	96	318	1500

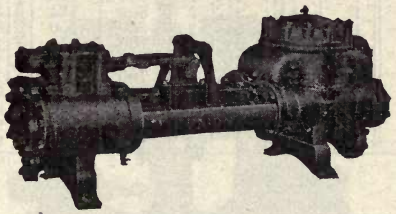


Fig. 103a

Union Horizontal Duplex Piston Pattern Boiler Feed or Pressure Pumps

250 Pounds Maximum Steam Pressure, 250 Pounds Maximum Water Pressure.

SIZE	Diam. Pump Openings For long pipe lines use larger pipes, reducing size at the pump openings				Ratings			
	Steam	Exhaust	Suction	Discharge	Gallons per Stroke, Each Cylinder	Max. Strokes, Each Cylinder per Minute	Capacity, G. P. M., Both Cylinders Max. Speed	H. P. Boilers Pump will feed at Slow Speed
2½x 1½x 3	⅜	½	1	¾	.023	160	7.36	40
3 x 2 x 3	⅜	½	1¼	1	.041	160	13.1	70
4½x 2¾x 4	½	¾	2	1½	.103	150	30	150
5¼x 3½x 5	¾	1	2½	1½	.208	140	58	250
6 x 4 x 6	1	1¼	3	2	.326	130	84	350
7½x 5 x 6	1½	2	4	3	.510	130	133	550
7½x 5 x 8	1½	2	4	3	.679	110	149	700
7½x 4½x10	1½	2	4	3	.689	96	132	700
9 x 5¼x10	1½	2	4	3	.938	96	180	950
10 x 6 x10	1½	2	5	4	1.224	96	235	1200
10 x 6 x12	1½	2	5	4	1.46	90	265	1500
10 x 7 x10	1½	2	6	5	1.66	96	318	1700
12 x 7 x12	2	2½	6	5	2.00	90	360	2000
12 x 8½x12	2	2½	6	5	2.94	90	530	3000
14 x 8½x12	2	2½	6	5	2.94	90	530	3000
14 x10 x12	2	2½	8	6	4.08	90	730	4150
16 x10 x12	2½	3	8	6	4.08	90	730	4150

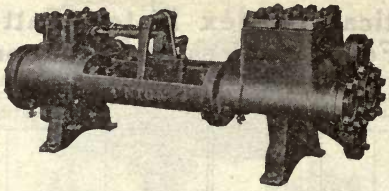


Fig. 181

Union Duplex Pressure Oil Pumps

250 Lbs. Max. Steam Pressure. Max. Oil Pressure Shown in Table

Size of Pump			Diameter of Pump Openings				Ratings				
Diameter of Steam Cyls.	Diameter of Oil Cylinders	Length of Stroke	Steam	Exhaust	Suction	Discharge	Max. Oil Pressure Lbs. Sq. Inch	Gallons Per Stroke Each Cyl.	Max. Strokes Per Minute Each Cyl.	Gallons Per Min. Both Cylinders Max. Speed	Bbls. Per Hr. (42 Gallons Per Barrel)
6	2½	6	1	1¼	2½	2	500	.128	130	33	47
6	3	6	1	1¼	2½	2	500	.184	130	38	54
6	3½	6	1	1¼	2½	2	500	.250	130	65	93
7½	2½	8	1½	2	2½	2	700	.170	110	37½	53
7½	3	8	1½	2	2½	2	700	.245	110	54	77
7½	3½	8	1½	2	2½	2	700	.333	110	73	104
10	3	12	1½	2	3	2	1000	.367	90	66	94
10	3½	12	1½	2	3	2	1000	.500	90	90	128
10	4	12	1½	2	4	3	800	.652	90	117	167
10	4½	12	1½	2	4	3	800	.826	90	148	210

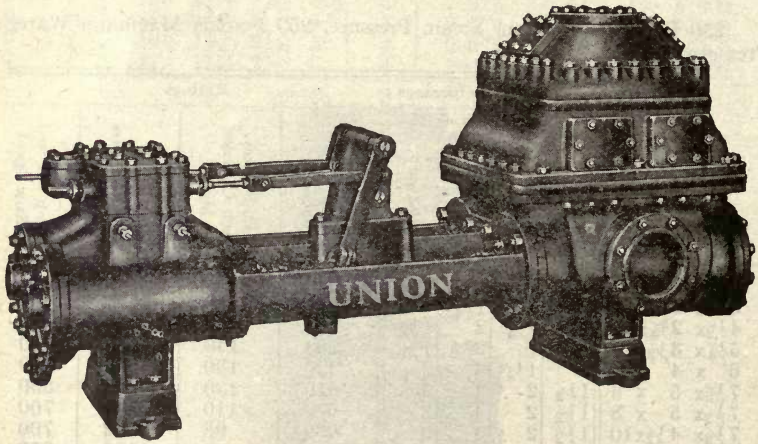


Fig. 205.

Union Duplex Oil Pumps

Separate-Chest Pattern.

200 Pounds Maximum Steam and Oil Pressure.

Size of Pump			Diameter of Pump Openings				Ratings			
Diameter Steam Cyls.	Diameter Oil Cyl.	Length of Stroke	Steam	Exhaust	Suction	Discharge	Gallons Per Stroke, Each Cyl.	Maximum Strokes Each Cyl. Per Min.	Gallons Per Min. Both Cyls.	Barrels (42 Gals.) Per Hour
10	8	18	2	2½	6	5	3.91	65	508	725
10	9	18	2	2½	8	6	4.95	65	643	914
10	10	18	2	2½	8	6	6.12	65	795	1135
12	8	18	2	2½	6	5	3.91	65	508	725
12	9	18	2	2½	8	6	4.95	65	643	914
12	10	18	2	2½	8	6	6.12	65	795	1135
14	8	18	2	2½	6	5	3.91	65	508	725
14	9	18	2	2½	8	6	4.95	65	643	914
14	10	18	2	2½	8	6	6.12	65	795	1135
16	12	18	2½	3	12	10	8.81	50	881	1258
16	14	18	2½	3	14	10	12.0	50	1200	1714
16	12	24	3	4	12	10	11.74	50	1174	1677
16	14	24	3	4	14	10	16.0	50	1600	2285

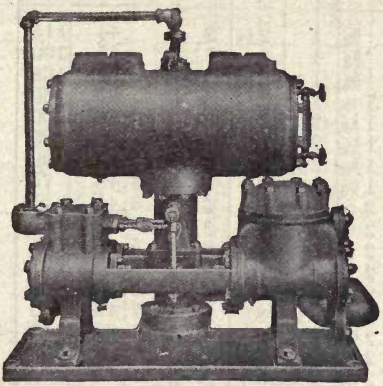


Fig. 183

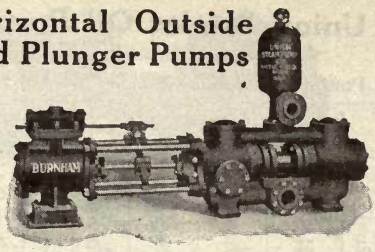
Union Duplex Automatic Feed Pumps and Receivers

250 Lbs. Maximum Steam Pressure. 250 Lbs. Maximum Water Pressure.

Receivers suitable for 150 Lbs. Maximum Pressure.

Size of Pump			Diameter of Pump Openings					Ratings	
			For long pipe lines, use larger pipes reducing size at the pump openings						
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Receiver Inlets, No. & Size	Gallons per Min. at Speed Recommended	Square Feet of Radiating Surface
2½	1½	3	⅜	½	1	¾	1–2″	2.75	2750
3	2	3	⅜	½	1¼	1	1–2″	5	5000
4½	2¾	4	½	¾	2	1½	1–2″	12	12000
5¼	3½	5	¾	1¼	2½	1½	2–3″	25	25000
6	4	6	1	1¼	3	2	2–3″	40	40000
7½	5	6	1½	2	4	3	2–3″	60	60000
7½	4½	10	1½	2	4	3	2–3″	70	70000
9	5¼	10	1½	2	4	3	2–3″	95	95000
10	6	10	1½	2	5	4	2–6″	120	120000
12	7	12	2	2½	6	5	2–6″	200	200000
14	8½	12	2	2½	8	6	2–6″	300	300000

Burnham Horizontal Outside Center-Packed Plunger Pumps



250 Pounds Maximum Steam and Water Pressures

Fig. 105a.

Size of Pump			Diam. Pump Openings For long pipe lines use larger pipes, reducing size at pump openings				RATINGS			
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Gallons per Stroke	Maximum Strokes per Minute	Gallons per Minute at Maximum Speed	Horse-Power of Boiler Pump will Feed at Slow Speed
6 1/8	4	8	3/4	1	2 1/2	2	.44	115	50.7	220
6 1/8	4	12	3/4	1	3	2 1/2	.65	100	65	283
7	4 1/2	10	3/4	1	3	2 1/2	.688	108	74.1	350
8	5	12	1	1 1/2	3 1/2	3	1.01	100	101	440
10	6	12	1 1/4	2	4	3	1.46	100	146	635
12	7	12	1 1/2	2 1/2	5	4	2.	100	200	870
12	8	12	1 1/2	2 1/2	5	4	2.61	100	261	1135
12	8	16	1 1/2	2 1/2	6	5	3.48	75	261	1510
14	8	16	2	2 1/2	6	5	3.48	75	261	1510
14	9	16	2	2 1/2	6	5	4.40	75	330	1920
16	10	16	2	2 1/2	8	6	5.44	75	400	2375
16	10 1/2	16	2	2 1/2	8	6	6.	75	450	2600
16	10	20	2	2 1/2	8	6	6.80	60	400	2950
16	10 1/2	20	2	2 1/2	8	6	7.49	60	450	3250
18	12	16	2 1/2	3 1/2	8	6	7.82	75	587	3400
18	12	20	2 1/2	3 1/2	8	6	9.78	60	587	4250
20	14	20	2 1/2	3 1/2	10	8	13.32	60	800	5800
24	16	20	3	5	10	8	17.4	60	1044	7500
26	18	20	3	5	12	10	22.	60	1320	9600
26	18	24	3	5	12	10	26.42	50	1320	11500
30	20	24	4	6	14	12	32.64	50	1632	14200

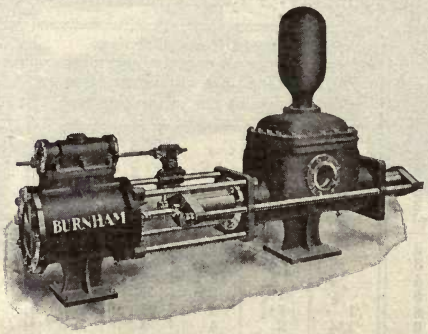


Fig. 107a

Burnham Horizontal Outside End Packed Plunger Pumps

In the sizes listed below, the 14x8x12 and smaller are suitable for a Maximum Working Pressure of 250 Pounds Steam and Water. Larger size sare suitable for a Maximum Working Pressure of 150 Pounds Steam and Water.

Size of Pump			Diam. Pump Openings				RATINGS			
			For long pipe lines use larger pipes, reducing size at pump openings							
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Gallons per Stroke	Maximum Strokes per Minute	Gallons per Minute at Maximum Speed	Horse-power of Boiler will feed at Slow Speed
4	2½	5	½	¾	1½	1¼	.106	140	14.8	62
5	3	6	½	¾	2	1½	.183	130	23.8	106
5½	3½	7	½	¾	2½	2	.291	120	35	147
6⅛	4	8	¾	1	2½	2	.44	115	50.7	220
7	4½	10	¾	1	3	2½	.688	108	74.1	350
8½	5	10	1	1½	3½	3	.85	108	92	430
10	6	12	1¼	2	4	3	1.46	100	146	635
12	7	12	1½	2½	5	4	2.00	100	200	870
12	8	12	1½	2½	5	4	2.61	100	261	1135
14	8	12	2	2½	5	4	2.61	100	261	1135
12	8	16	1½	2½	6	5	3.48	75	261	1510
14	8	16	2	2½	6	5	3.48	75	261	1510
14	9	16	2	2½	6	5	4.40	75	330	1920
16	10	16	2	2½	8	6	5.44	75	400	2375
18	12	16	2½	3½	8	6	7.82	75	587	3400
20	14	16	2½	3½	10	8	10.66	75	800	4650
24	16	20	3	5	10	8	17.40	60	1044	7500
26	18	20	3	5	12	10	22.00	60	1320	9600

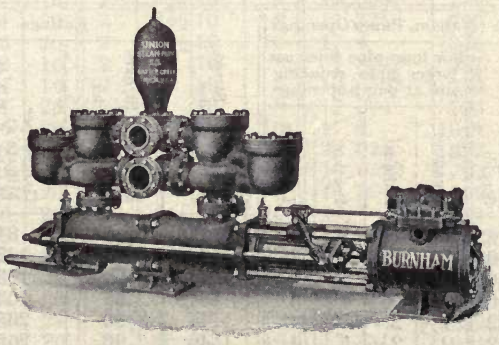


Fig 109a

Pot-Valve Plunger Pump.

Burnham Horizontal Outside End Packed Pot Valve Plunger Pumps

250 Pounds Maximum Steam and Water Pressures.

Size of Pump			Diam. Pump Openings				Ratings			
			For long pipe lines use larger pipes, reducing size at pump Openings							
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Gallons per Stroke	Maximum Strokes per Minute	Gallons per Minute at Maximum Speed	Horse Power of Boiler Pump will Feed at Slow Speed
5½	3	7	½	¾	2½	2	.214	120	25.7	105
5½	3½	7	½	¾	2½	2	.291	120	35	147
6⅓	4	8	¾	1	2½	2	.44	115	50.6	220
7	4½	10	¾	1	3	2½	.688	108	74.3	350
7	5	10	¾	1	3½	3	.85	108	92	430
8	5	12	1	1½	3½	3	1.01	100	101	440
8	5½	12	1	1½	3½	3	1.23	100	123	530
10	6	12	1¼	2	4	3	1.46	100	146	635
12	7	16	1½	2½	5	4	2.66	75	200	1160
12	7½	16	1½	2½	5	4	3.05	75	229	1330
14	8	16	2	2½	6	5	3.48	75	261	1510
14	9	16	2	2½	6	5	4.40	75	330	1920
16	10	16	2	2½	8	6	5.44	75	400	2375
16	10½	16	2	2½	8	6	6.00	75	450	2600
16	10	20	2	2½	8	6	6.80	60	400	2950
16	10½	20	2	2½	8	6	7.49	60	450	3250

Union Horizontal Duplex Outside End Packed Pot Valve Plunger Pumps

250 Pounds Max. Steam Pressure,
300 Pounds Max. Water Pressure.

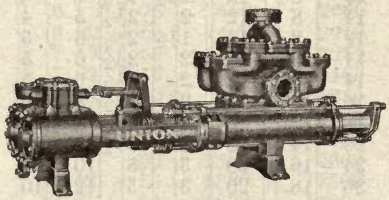


Fig. 203.

Size of Pump			Diam. Pump Openings				Ratings			
			For long pipe lines use larger pipes, reducing size at pump openings							
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Gallons per Stroke Each Cylinder	Maximum Strokes per Minute Each Cylinder	Gallons per Minute at Maximum Speed	Horse-Power of Boiler Pump will Feed at Slow Speed
6	4	6	1	1¼	3	2	.326	130	84	350
7½	4½	10	1½	2	4	3	.689	96	132	700
9	5	10	1½	2	4	3	.819	96	163	860
10	5	10	1½	2	4	3	.849	96	163	860
10	6	12	1½	2	5	4	1.46	90	265	1500
12	6	12	2	2½	5	4	1.46	90	265	1500
12	7	12	2	2½	6	5	2.00	90	360	2000
14	7	12	2	2½	6	5	2.00	90	360	2000

Burnham Compound Steam Pumps

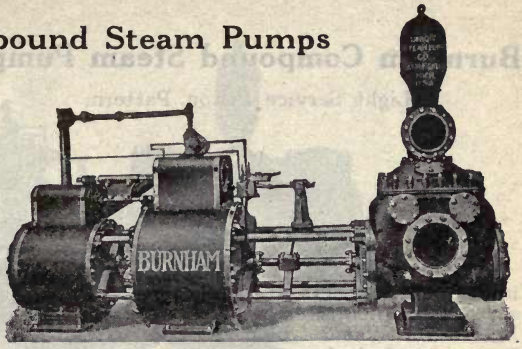


Fig. 117.

Heavy Service Piston Pattern

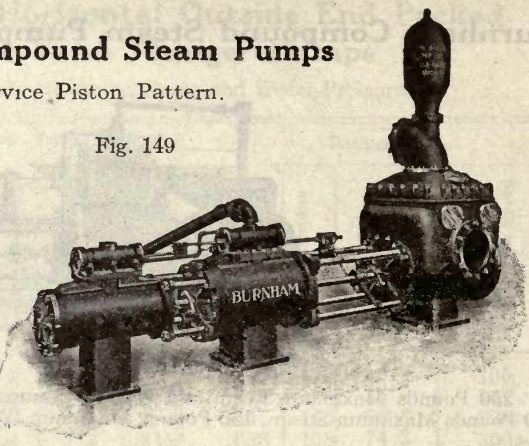
250 Pounds Maximum Steam and Water Pressures for first five sizes. 200 Pounds Maximum Steam, 150 Pounds Maximum Water Pressures larger sizes.

Size of Pump				Diam. Pump Openings				Ratings		
				For long pipe lines use larger pipes, reducing size at the pump openings						
Diameter of High-Pressure Steam Cylinder	Diameter of Low-Pressure Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Displacement in Gallons per Stroke	Gallons per Minute at 100 feet Piston Speed	Gallons per Hour at Given Speed
8	12	6	12	1	2½	4	3	1.4676	146	8812
8	12	7	12	1	2½	5	4	1.998	200	11995
8	12	8	12	1	2½	5	4	2.61	261	15667
10	16	9	16	1¼	2½	6	5	4.4048	330	19828
10	16	10	16	1¼	2½	8	6	5.44	408	24480
12	18	9	16	1½	3½	6	5	4.4048	330	19828
12	18	10	16	1½	3½	8	6	5.44	408	24480
12	18	12	16	1½	3½	8	6	7.8272	587	35251
12	18	14	16	1½	3½	10	8	10.6592	799	47980
12	18	14	20	1½	3½	10	8	13.324	799	47980
14	20	10	16	2	3½	8	6	5.44	408	24480
14	20	12	16	2	3½	8	6	7.8272	587	35251
14	20	14	16	2	3½	10	8	10.6592	799	47980
14	20	14	20	2	3½	10	8	13.324	799	47980
14	20	16	20	2	3½	12	10	17.4	1044	62668
16	24	12	16	2	5	10	8	7 8272	587	35251
16	24	14	16	2	5	10	8	10.6592	799	47980
16	24	14	20	2	5	10	8	13.324	799	47980
16	24	16	20	2	5	12	10	17.4	1044	62668
16	24	18	20	2	5	12	10	22.024	1322	79314
18	26	14	16	2½	5	10	8	10.6592	799	47980
18	26	14	20	2½	5	10	8	13.324	799	47980
18	26	16	20	2½	5	12	10	17.4	1044	62668
18	26	18	20	2½	5	12	10	22.024	1322	79314
18	26	20	24	2½	5	14	12	32.64	1632	97920
20	30	16	20	2½	5	12	10	17.4	1044	62668
20	30	18	20	2½	5	12	10	22.024	1322	79314
20	30	20	24	2½	6	14	12	32.64	1632	97920
24	36	16	20	3	6	12	10	17.4	1044	62668
24	36	18	20	3	6	12	10	22 024	1322	79314
24	36	20	24	3	6	14	12	32.64	1632	97920

Burnham Compound Steam Pumps

Light Service Piston Pattern.

Fig. 149



200 Pounds Maximum Steam Pressure, 100 Pounds Maximum Water Pressure, 12 inch stroke and smaller, 75 lbs. Maximum Water Pressure for larger sizes.

Size of Pump				Diam. Pump Openings (For long pipe lines use larger pipes, reducing size at the pump openings)				Ratings		
Diameter High-Pressure Steam Cylinder	Diameter Low-Pressure Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Displacement in Gallons per Stroke	Gallons per Minute at 100 Feet Piston Speed	Gallons per Hour at Given Speed
5	8½	6	10	¾	1½	4	3½	1.22	146	8812
5	8½	7	10	¾	1½	5	4	1.665	200	11995
5	8½	8	10	¾	1½	5	4	2.175	261	15667
5	8½	10	10	¾	1½	6	5	3.40	408	24480
6⅛	8½	7	10	¾	1½	5	4	1.665	200	11995
6⅛	8½	8	10	¾	1½	5	4	2.175	261	15667
6⅛	8½	10	10	¾	1½	6	5	3.40	408	24480
8	12	10	12	1	2½	8	6	4.08	408	24480
8	12	12	16	1	2½	8	6	7.82	587	35251
8	12	14	16	1	2½	10	8	10.65	799	47980
10	16	12	16	1¼	2½	8	6	7.82	587	35251
10	16	14	16	1¼	2½	10	8	10.65	799	47980
10	16	16	16	1¼	2½	10	8	13.92	1044	62668
12	18	14	16	1½	3½	10	8	10.65	799	47980
12	18	16	16	1½	3½	10	8	13.92	1044	62668
12	18	16	20	1½	3½	12	10	17.40	1044	62668
12	18	18	20	1½	3½	12	10	22.02	1322	79314
14	20	16	20	2	3½	12	10	17.40	1044	62668
14	20	18	20	2	3½	12	10	22.02	1322	79314
14	20	20	24	2	3½	14	12	32.64	1632	97920
14	20	22	24	2	3½	15	12	39.48	1974	118482
16	24	20	24	2	5	14	12	32.64	1632	97920
16	24	22	24	2	5	15	12	39.48	1974	118482
16	24	24	24	2	5	16	14	46.99	2349	140958
18	26	20	24	2½	5	14	12	13.64	1632	97920
18	26	22	24	2½	5	15	12	39.48	1974	118482
18	26	24	24	2½	5	16	14	46.99	2349	140958
20	30	22	24	2½	5	15	12	39.48	1974	118482
24	36	26	24	3	6	18	16	55.15	2758	165484
24	36	28	24	3	6	18	16	63.96	3199	191922

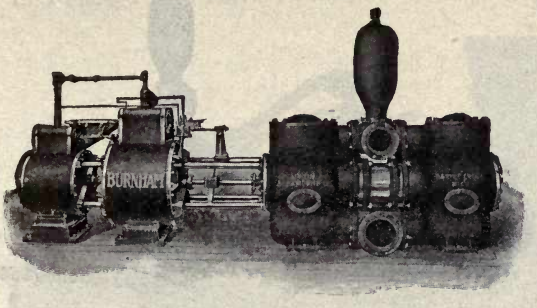


Fig. 150

Burnham Compound Steam Pumps

Outside-Center-Packed Plunger Pattern

200 Pounds Maximum Steam Pressure, 250 Pounds Maximum Water Pressure.

Size of Pump				Diameter of Pump Openings				Ratings		
				For long pipe lines use larger pipes, reducing size at the pump openings						
Diameter of High-Presure Steam Cylinder	Diameter of Low-Presure Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Displacement in Gallons per Stroke	Gallons per Minute at 100 feet Piston Speed	Gallons per Hour at Given Speed
8	12	6	12	1	2½	4	3	1.4676	146	8812
8	12	7	12	1	2½	5	4	1.998	200	11995
8	12	8	16	1	2½	6	5	3.48	261	15667
10	16	8	16	1¼	2½	6	5	3.48	261	15667
10	16	9	16	1½	2½	6	5	4.4048	330	19828
10	16	10	16	1¼	2½	8	6	5.44	408	24480
12	18	9	16	1½	3½	6	5	4.4048	330	19828
12	18	10	16	1½	3½	8	6	5.44	408	24480
12	18	12	16	1½	3½	8	6	7.8272	587	35251
12	18	14	20	1½	3½	10	8	13.324	799	47980
14	20	10	16	2	3½	8	6	5.44	408	24480
14	20	12	16	2	3½	8	6	7.8272	587	35251
14	20	14	20	2	3½	10	8	13.324	799	47980
16	24	12	16	2	5	8	6	7.8272	587	35251
16	24	14	20	2	5	10	8	13.324	799	47980
16	24	16	20	2	5	10	8	17.4	1044	62668
18	26	14	20	2½	5	10	8	13.324	799	47980
18	26	16	20	2½	5	10	8	17.4	1044	62668
18	26	18	24	2½	5	12	10	26.4288	1322	79314
20	30	16	20	2½	5	10	8	17.4	1044	62668
20	30	18	24	2½	5	12	10	26.4288	1322	79314
20	30	20	24	2½	5	14	12	32.64	1632	97920
24	36	16	20	3	6	10	8	17.4	1044	62668
24	36	18	24	3	6	12	10	26.4288	1322	79314
24	36	20	24	3	6	14	12	32.64	1632	97920

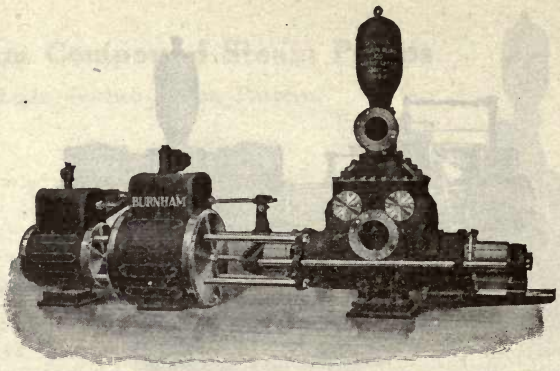


Fig. 151

Burnham Compound Steam Pumps

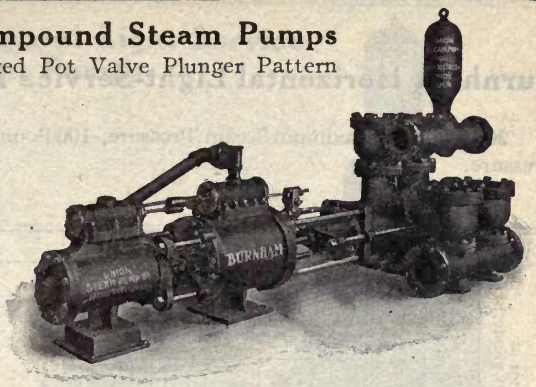
Outside End-Packed Plunger Pattern

250 Pounds Maximum Steam and Water Pressures for 12 inch stroke pumps, 150 Pounds Maximum Steam and Water Pressures for larger sizes

Size of Pump				Diameter of Pump Openings				Ratings		
				For long pipe lines use larger pipes, reducing size at the pump openings						
Diameter of High-Pressure Steam Cylinder	Diameter of Low-Pressure Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Displacement in Gallons per Stroke	Gallons per Minute at 100 feet Piston Speed	Gallons per Hour at Given Speed
8	12	6	12	1	2½	4	3	1.4676	146	8812
8	12	7	12	1	2½	5	4	1.998	200	11995
8	12	8	12	1	2½	5	4	2.61	261	15667
10	16	8	16	1¼	2½	6	5	3.48	261	15667
10	16	9	16	1¼	2½	6	5	4.4048	330	19828
10	16	10	16	1¼	2½	8	6	5.44	408	24480
12	18	9	16	1½	3½	6	5	4.4048	330	19828
12	18	10	16	1½	3½	8	6	5.44	408	24480
12	18	12	16	1½	3½	8	6	7.8272	587	35251
12	18	14	16	1½	3½	10	8	10.6592	799	47980
14	20	10	16	2	3½	8	6	5.44	408	24480
14	20	12	16	2	3½	8	6	7.8272	587	35251
14	20	14	16	2	3½	10	8	10.6592	799	47980
16	24	12	16	2	5	8	6	7.8282	587	35251
16	24	14	16	2	5	10	8	10.6592	799	47980
16	24	16	20	2	5	12	10	17.4	1044	62668
16	24	18	20	2	5	12	10	22.024	1322	79314
18	26	14	16	2½	5	10	8	10.6592	799	47980
18	26	16	20	2½	5	12	10	17.4	1044	62668
18	26	18	20	2½	5	12	10	22.024	1322	79314
20	30	16	20	2½	5	12	10	17.4	1044	62668
20	30	18	20	2½	5	12	10	22.024	1322	79314
24	36	16	20	3	5	12	10	17.4	1044	62668
24	36	18	20	3	5	12	10	22.024	1322	79314

Burnham Compound Steam Pumps

Outside End-Packed Pot Valve Plunger Pattern

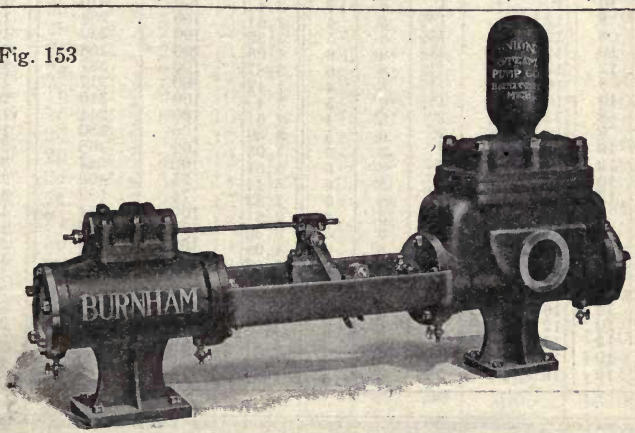


250 Pounds Maximum Steam and Water Pressures.

Fig. 152

Size of Pump				Diam. Pump Openings (For long pipe lines use larger pipes, reducing size at the pump openings)				Ratings		
Diameter of High-Pressure Steam Cylinder	Diameter of Low-Pressure Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Displacement in Gallons per Stroke	Gallons per Minute at 100 feet Piston Speed	Gallons per Hour at Given Speed
8	12	6	12	1	2½	4	3	1.4676	146	8812
8	12	7	16	1	2½	5	4	2.664	200	11995
8	12	7½	16	1	2½	5	4	3.072	229	13769
10	16	8	16	1¼	2½	6	5	3.48	261	15667
10	16	9	16	1¼	2½	6	5	4.4048	330	19828
10	16	10	16	1¼	2½	8	6	5.44	408	24480
10	16	10½	16	1¼	2½	8	6	6.01	449	26989
12	18	9	16	1½	3½	6	5	4.4048	330	19828
12	18	10	16	1½	3½	8	6	5.44	408	24480
12	18	10½	16	1½	3½	8	6	6.01	449	26989
12	18	10	20	1½	3½	8	6	6.8	408	24480
12	18	10½	20	1½	3½	8	6	7.513	449	26989
14	20	10	16	2	3½	8	6	5.44	408	24480
14	20	10½	16	2	3½	8	6	6.01	449	26989
14	20	10	20	2	3½	8	6	6.8	408	24480
14	20	10½	20	2	3½	8	6	7.513	449	26989

Fig. 153



Burnham Horizontal Light-Service Piston Pumps

200 Pounds Maximum Steam Pressure, 100 Pounds Maximum Water Pressure.

Size of Pump			Diameter of Pump Openings				Ratings			
			For long pipe lines use larger pipes, reducing size at pump opening							
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Gallons per Stroke	Strokes per Minute	Gallons per Minute at Given Number of Strokes	Gallons per Hour at Given Speed
3	2½	3	⅜	½	1¼	1	.06	150	9.5	570
3	3	3	⅜	½	1½	1¼	.09	150	13.8	828
3	3½	3	⅜	½	1½	1¼	.12	150	18.7	1122
4	3	5	½	¾	2	1½	.15	140	21.4	1264
4	4	5	½	¾	2½	2	.27	140	38	2280
4	4½	5	½	¾	3	2½	.34	140	48	2880
4	5	5	½	¾	3½	3	.42	140	59.5	3570
4	5½	5	½	¾	3½	3	.51	140	72	4320
4	6	5	½	¾	4	3½	.61	140	85.5	5130
5	3½	6	½	¾	2½	2	.24	125	31	1860
5	4	6	½	¾	2½	2	.32	125	40.5	2430
5	4½	6	½	¾	3	2½	.41	125	51.5	3090
5	5	6	½	¾	3½	3	.50	125	63.5	3810
5	6	6	½	¾	4	3½	.73	125	92	5520
5	5	10	½	¾	3½	3	.84	100	85	5100
5	6	10	½	¾	4	3½	1.22	100	122	7320
5	7	10	½	¾	5	4	1.66	100	166	9960
5	8	10	½	¾	5	4	2.17	100	217	13020
5½	4	7	½	¾	2½	2	.38	125	47.5	2850
5½	5	7	½	¾	3½	3	.59	125	74.5	4470
5½	6	7	½	¾	4	3½	.85	125	107	6420
5½	7	7	½	¾	5	4	1.16	125	146	8760
6⅛	5	8	¾	1	3½	3	.67	125	85	5100
6⅛	6	8	¾	1	4	3½	.97	125	122	7320
6⅛	7	8	¾	1	5	4	1.33	125	166	9960
6⅛	6	10	¾	1	4	3½	1.22	100	122	7320
6⅛	7	10	¾	1	5	4	1.66	100	166	9960
6⅛	8	10	¾	1	5	4	2.17	100	217	13020
6⅛	9	10	¾	1	6	5	2.75	100	275	16500
6⅛	10	10	¾	1	6	5	3.40	100	340	14400
7	6	10	¾	1	4	3½	1.22	100	122	7320
7	7	10	¾	1	5	4	1.66	100	166	9960
7	8	10	¾	1	5	4	2.17	100	217	13020
7	9	10	¾	1	6	5	2.75	100	275	16500
7	10	10	¾	1	6	5	3.4	100	340	20400
8	8	12	1	1½	5	4	2.61	100	261	15600
8	9	12	1	1½	6	5	3.30	100	330	19800
8	10	12	1	1½	8	6	4.08	100	408	24480
8	12	12	1	1½	8	6	5.87	100	587	35220
8½	8	10	1	1½	5	4	2.17	100	217	13020
8½	10	10	1	1½	6	5	3.4	100	340	20400
10	8	12	1¼	2	5	4	2.61	100	261	15660
10	9	12	1¼	2	6	5	3.30	100	330	19800
10	10	12	1¼	2	8	6	4.08	100	408	24480
10	12	12	1¼	2	8	6	5.87	100	587	35220
10	10	16	1¼	2	8	6	5.44	75	408	24480
10	12	16	1¼	2	8	6	7.82	75	587	35220
12	8	12	1½	2½	5	4	2.61	100	261	15660
12	9	12	1½	2½	6	5	3.30	100	330	19800
12	10	12	1½	2½	8	6	4.08	100	408	24480
12	12	12	1½	2½	8	6	5.87	100	587	35220
12	9	16	1½	2½	6	5	4.40	75	330	19800
12	10	16	1½	2½	8	6	5.44	75	408	24480
12	12	16	1½	2½	8	6	7.82	75	587	35220
14	10	12	2	2½	8	6	4.08	100	408	24480
14	12	12	2	2½	8	6	5.87	100	587	35220
14	10	16	2	2½	8	6	5.44	75	408	24480
14	12	16	2	2½	8	6	7.82	75	587	35220

Burham Horizontal Light-Service Piston Pumps

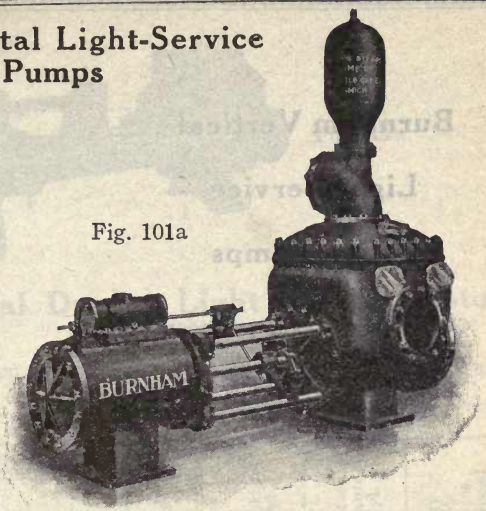


Fig. 101a

200 Pounds Maximum Steam Pressure, 75 Pounds Maximum Water Pressure.

Size of Pump			Diameter of Pump Openings				Ratings			
			For long pipe lines use larger pipes, reducing size at the pump openings							
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Gallons per Stroke	Strokes per Minute	Gallons per Minute at Given Number of Strokes	Gallons per Hour at Given Speed
10	12	16	1¼	2	8	6	7.82	75	587	35220
10	14	16	1¼	2	10	8	10.65	75	800	48000
10	16	16	1¼	2	10	8	13.92	75	1042	62520
12	12	16	1½	2½	8	6	7.82	75	587	35220
12	14	16	1½	2½	10	8	10.65	75	800	48000
12	16	16	1½	2½	10	8	13.92	75	1042	62520
14	12	16	2	2½	8	6	7.82	75	587	35220
14	14	16	2	2½	10	8	10.65	75	800	48000
14	16	16	2	2½	10	8	13.92	75	1042	62520
14	16	20	2	2½	10	8	17.4	60	1044	62640
14	18	20	2	2½	12	10	22.02	60	1320	79200
16	14	16	2	2½	10	8	10.65	75	800	48000
16	16	16	2	2½	10	8	13.92	75	1042	62520
16	16	20	2	2½	10	8	17.4	60	1044	62640
16	18	20	2	2½	12	10	22.02	60	1320	79200
16	20	24	2	2½	14	12	32.64	50	1632	99720
16	22	24	2	2½	15	12	39.48	50	1974	118440
18	14	16	2½	3½	10	8	10.65	75	800	48000
18	16	16	2½	3½	10	8	13.92	75	1042	62520
18	16	20	2½	3½	10	8	17.4	60	1044	62640
18	18	20	2½	3½	12	10	22.02	60	1320	79200
18	20	24	2½	3½	14	12	32.64	50	1632	97920
18	22	24	2½	3½	15	12	39.48	50	1974	118440
18	24	24	2½	3½	16	14	46.99	50	2350	141000
20	16	20	2½	3½	10	8	17.4	60	1044	62640
20	18	20	2½	3½	12	10	22.02	60	1320	79200
20	20	24	2½	3½	14	12	32.64	50	1632	97920
20	22	24	2½	3½	15	12	39.48	50	1974	118440
20	24	24	2½	3½	16	14	46.99	50	2350	141000
20	26	24	2½	3½	18	16	55.15	50	2757	165420
20	28	24	2½	3½	18	16	63.96	50	3198	191880
24	24	24	3	5	16	14	46.99	50	2350	141000
24	26	24	3	5	18	16	55.15	50	2757	165420
24	28	24	3	5	18	16	63.96	50	3198	191880

Burnham Vertical Light-Service Piston Pumps

200 Pounds Maximum Steam Pressure, 100 Pounds Maximum Water Pressure.

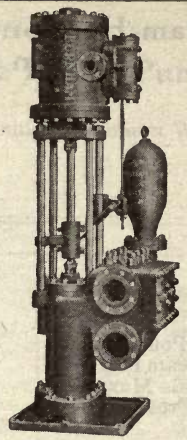


Fig. 112a

Size of Pump			Diam. Pump Openings				Ratings			
			For long pipe lines use larger pipes, reducing size at the pump openings							
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Gallons per Stroke	Strokes per Minute	Gallons per Minute at Given Number of Strokes	Gallons per Hour at Given Speed
4	3	5	½	¾	2½	2	.15	140	21.4	1264
5	3½	6	½	¾	2½	2	.24	125	31	1860
5½	4	7	½	¾	3	2½	.38	125	47.5	2850
5½	4½	7	½	¾	3	2½	.48	125	60	3600
6⅛	5	10	¾	1	3½	3	.84	100	85	5100
6⅛	6	10	¾	1	4	3	1.22	100	122	7320
7	5	10	¾	1	3½	3	.84	100	85	5100
7	6	10	¾	1	4	3	1.22	100	122	7320
8	6	12	1	1½	4	3	1.46	100	146	8780
8	7	12	1	1½	5	4	2.00	100	200	12000
10	7	12	1¼	2	5	4	2.00	100	200	12000
10	8	12	1¼	2	5	4	2.61	100	261	15600
12	8	12	1½	2½	5	4	2.61	100	261	15600
10	9	16	1¼	2	6	5	4.4	75	330	19800
10	10	16	1¼	2	6	5	5.44	75	408	24480
12	9	16	1½	2½	6	5	4.4	75	330	19800
12	10	16	1½	2½	6	5	5.44	75	408	24480
14	10	16	2	2½	6	5	5.44	75	408	24480

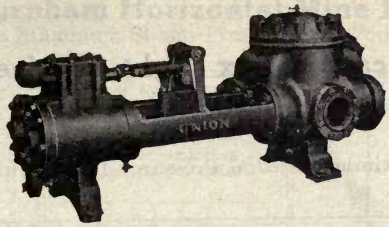


Fig. 102a

Union Horizontal Duplex Light Service Piston Pumps

250 Pounds Maximum Steam Pressure, 125 Pounds Maximum Water Pressure.

SIZE	Diam. of Pump Openings (For long pipe lines use larger pipes, reducing size at the pump openings)				Ratings		
	Steam	Exhaust	Suction	Discharge	Gallons per Stroke, Each Cyl.	Max. Strokes per Minute, Each Cyl.	Capacity G. P. M. Both Cyls., Max. Speed
4¼x 3¾x 4	½	¾	2½	1½	.191	150	57
5¼x 4¾x 5	¾	1	3	2	.384	140	107
6 x 5¾x 6	1	1¼	4	3	.674	130	175
6 x 6 x 6	1	1¼	4	3	.732	130	190
7½x 6 x 6	1½	2	4	3	.732	130	190
7½x 7½x 6	1½	2	6	5	1.15	130	298
7½x 6 x10	1½	2	5	4	1.22	96	235
7½x 7 x10	1½	2	6	5	1.66	96	318
7½x 8½x10	1½	2	6	5	2.45	96	470
9 x 8½x10	1½	2	6	5	2.45	96	470
10 x 8½x10	1½	2	6	5	2.45	96	470
10 x10 x10	1½	2	8	6	3.39	96	650
12 x10¼x12	2	2½	8	6	4.29	90	773
14 x10¼x12	2	2½	8	6	4.29	90	773
12 x12 x12	2	2½	10	8	5.87	90	1058
14 x12 x12	2	2½	10	8	5.87	90	1058
16 x12 x12	2½	3	10	8	5.87	90	1058

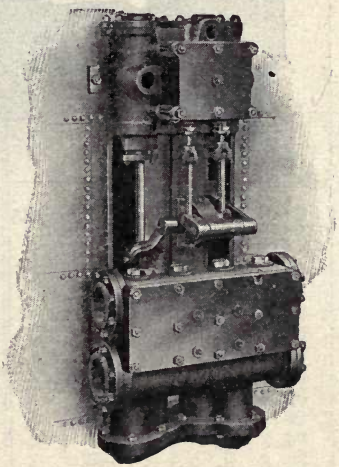


Fig. 114

Union Vertical Duplex Light Service Piston Pumps

200 Pounds Maximum Steam Pressure, 100 Pounds Maximum Water Pressure.

SIZE	OPENINGS				Ratings		
	Steam	Exhaust	Suction	Discharge	Gallons per Stroke Each Cyl.	Max. Strokes per Minute Each Cyl.	Capacity G. P. M. both Cyls. Max. Speed
4½x 3¾x 4	½	¾	2½	1½	.191	150	57
5¼x 4¾x 5	¾	1¼	3	2	.384	140	107
6 x 6 x 6	1	1¼	4	3	.732	130	190
7½x 6 x 6	1½	2	4	3	.732	130	190
7½x 7½x 6	1½	2	6	5	1.15	130	298
7½x 6 x10	1½	2	6	5	1.22	108	264
7½x 9 x10	1½	2	6	5	2.75	108	595
7½x10¼x10	1½	2	8	7	3.57	108	775
10 x12 x12	2	2½	8	7	5.89	90	1050

Fig. 134

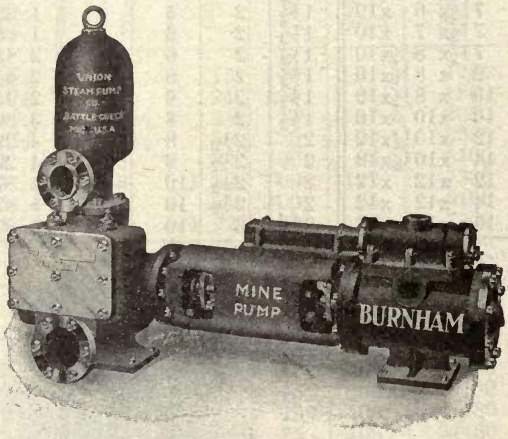


Fig. 135

Burnham Horizontal Mine Pumps (Fig. 135)

200 Pounds Maximum Steam Pressure, 150 Pounds Maximum Water Pressure.

Size of Pump			Diam. of Pump Openings				Ratings		
			For long pipe lines use larger pipes, reducing size at the pump openings						
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Gallons per Stroke	Gallons per Minute at 100 feet Piston Travel per Minute	Gallons per Hour
6	3	7	¾	1	2½	2	.21	36.7	2200
6	3½	7	¾	1	2½	2	.29	50	3000
6	4	7	¾	1	2½	2	.38	65	3900
7	3½	7	1	1½	2½	2	.29	50	3000
7	4	7	1	1½	2½	2	.38	65	3900
7	3½	12	1	1½	3	2½	.49	49	2940
7	4	12	1	1½	3	2½	.65	65	3900
7	5	13	1	1½	4	3	1.10	102	6120
8	5	13	1	1½	4	3	1.10	102	6120
10	5	13	1½	2½	4	3	1.10	102	6120
10	6	13	1½	2½	4	3½	1.59	148	8880
12	5	13	1½	2½	4	3	1.10	102	6120
12	6	13	1½	2½	4	3½	1.59	148	8880
12	7	13	1½	2½	5	4	2.16	200	12000
14	7	13	2	2½	5	4	2.16	200	12000
14	8	13	2	2½	6	5	2.82	260	15600
14	8	24	2	2½	6	5	5.22	260	15600
14	9	18	2	2½	6	5	4.95	331	19860
16	9	18	2	2½	6	5	4.95	331	19860
16	10	18	2	2½	8	6	6.12	410	24600
16	10½	18	2	2½	8	6	6.74	451	27060
18	10½	18	2½	3½	8	6	6.74	451	27060

Burnham Vertical Sinking Pumps (Fig. 134)

200 Pounds Maximum Steam Pressure, 150 Pounds Maximum Water Pressure.

Size of Pump			Diameter of Pump Openings				Ratings		
			For long pipe lines use larger pipes, reducing size at pump openings						
Diameter of Steam Cylinder	Diameter of Plunger	Length of Stroke	Steam	Exhaust	Suction	Discharge	Gallons per Stroke	Gallons per Minute	Gallons per Hour
7	3½	12	1	1½	2½	2	.5	50	3000
7	4	12	1	1½	3	2½	.65	65	3900
8	3½	12	1	1½	2½	2	.5	50	3000
8	4	12	1	1½	3	2½	.65	65	3900
10	5	13	1½	2½	4	3	1.10	100	6000
10	6	13	1½	2½	4	3½	1.58	150	9000
12	5	13	1½	2½	4	3	1.10	100	6000
12	6	13	1½	2½	4	3½	1.58	150	9000
12	7	13	1½	2½	5	4	2.16	200	12000
14	7	13	2	2½	5	4	2.16	200	12000
14	8	13	2	2½	6	5	2.82	261	15667
14	8½	13	2	2½	6	5	3.19	294	17688
16	8	16	2	2½	6	5	3.48	261	15667
16	8½	16	2	2½	6	5	3.93	294	17688

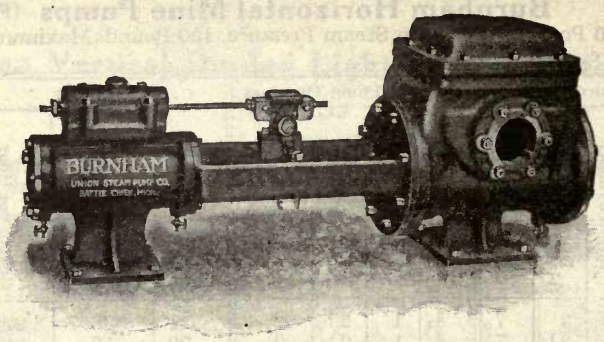


Fig. 141a
Low Vacuum Pump

Burnham Horizontal Low Vacuum Pumps

Standard Pattern—Cast-Yoke Pumps

200 Pounds Maximum Steam Pressure, 20 inch Vacuum with 30 inch Barometer.

Size of Pump			Ratings		SIZE OF OPENINGS For long pipe lines use larger pipes, reducing size at the pump openings			
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Square feet of Radiating Surface	Displacement in Gallons per Min. at 100 Strokes per Minute	Steam	Exhaust	Suction	Discharge
3	3½	4	2000	16	⅜	½	2	1½
3	4	4	2500	21	⅜	½	2	1½
4	3½	5	2500	21	½	¾	2	1½
4	4	5	3500	27	½	¾	2	1½
4	5	5	5000	42	½	¾	2½	2
4	6	5	7000	61	½	¾	3	2½
4	3½	6	3000	25	½	¾	2	1½
4	4	6	4000	32	½	¾	2	1½
4½	4	8	5000	43	½	¾	2	1½
4½	5	8	8000	68	½	¾	2½	2
4½	6	8	11000	98	½	¾	3	2½
4½	7	8	16000	133	½	¾	3½	3
5	5	6	6000	51	½	¾	2½	2
5	6	6	9000	73	½	¾	3	2½
5	7	6	12000	100	½	¾	3½	3
5	6	10	13000	122	½	¾	3	2½
5	7	10	19000	166	½	¾	3½	3
5	8	10	25000	217	½	¾	4	3½
5	8	12	33000	261	½	¾	4	3½
5½	6	7	10000	85	½	¾	3	2½
5½	7	7	13000	110	½	¾	3½	3
6⅛	6	8	11000	98	¾	1	3	2½
6⅛	7	8	16000	133	¾	1	3½	3
6⅛	8	8	21000	174	¾	1	4	3½
6⅛	6	10	13000	122	¾	1	3	2½
6⅛	7	10	19000	166	¾	1	3½	3
6⅛	8	10	25000	217	¾	1	4	3½
6⅛	9	10	32000	275	¾	1	4½	3½

Burnham Horizontal Low Vacuum Pumps (Continued)

Standard Pattern—Cast-Yoke Pumps

200 Pounds Maximum Steam Pressure, 20 inch Vacuum with 30 inch Barometer.

Size of Pump			Ratings		Size of Openings			
					For long pipe lines use larger pipes, reducing at the pump openings			
Diameter of Steam-Cylinder	Diameter of Water-Cylinder	Length of Stroke	Square feet of Radiating Surface	Displacement in Gallons per Min. at 100 Strokes per Minute	Steam	Exhaust	Suction	Discharge
6 1/8	10	10	38000	340	3/4	1	5	4
6 1/8	8	12	30000	261	3/4	1	4	3 1/2
6 1/8	10	12	45000	408	3/4	1	5	4
7	7	10	19000	166	3/4	1	3 1/2	3
7	8	10	25000	217	3/4	1	4	3 1/2
7	9	10	32000	275	3/4	1	4 1/2	3 1/2
7	10	10	38000	340	3/4	1	5	4
8	8	12	30000	261	1	1 1/2	4	3 1/2
8	9	12	38000	330	1	1 1/2	5	4
8	10	12	45000	408	1	1 1/2	5	4
8	12	12	65000	587	1	1 1/2	6	5
8	12	16	85000	782	1	1 1/2	6	5
8	14	16	110000	1065	1	1 1/2	7	6
8 1/2	8	10	25000	217	1	1 1/2	4	3 1/2
8 1/2	9	10	32000	275	1	1 1/2	4 1/2	3 1/2
8 1/2	10	10	38000	340	1	1 1/2	5	4
10	10	12	45000	408	1 1/4	2	5	4
10	12	12	65000	587	1 1/4	2	6	5
10	12	16	85000	782	1 1/4	2	6	5
10	14	16	110000	1065	1 1/4	2	7	6
12	10	12	45000	408	1 1/2	2 1/2	5	4
12	12	12	65000	587	1 1/2	2 1/2	6	5
12	12	16	85000	782	1 1/2	2 1/2	6	5
12	14	16	110000	1065	1 1/2	2 1/2	7	6

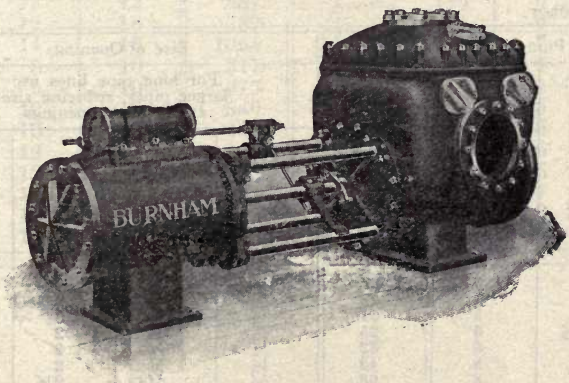


Fig. 141

Burnham Horizontal Low Vacuum Pumps

Standard Pattern—Rod-Yoke Pumps

200 Pounds Maximum Steam Pressure, 20 inch Vacuum with 30 inch Barometer.

Size of Pump			Ratings		Size of Openings			
					For long pipe lines use larger pipes, reducing size at the pump openings			
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Square Feet of Radiating Surface	Displacement in Gallons per Min. at 100 Strokes per Minute	Steam	Exhaust	Suction	Discharge
10	14	20	112000	1330	1½	2½	7	6
12	14	20	112000	1330	1½	2½	7	6
12	16	20	135000	1740	1½	2½	8	6
12	18	20	185000	2200	1½	2½	10	8
14	14	20	112000	1330	2	2½	7	6
14	16	20	135000	1740	2	2½	8	6
14	18	20	185000	2200	2	2½	10	8
14	20	24	250000	3264	2	2½	10	8
16	16	20	135000	1740	2	2½	8	6
16	18	20	185000	2200	2	2½	10	8
16	20	24	250000	3264	2	2½	10	8
18	18	20	185000	2200	2½	3½	10	8
18	20	24	250000	3264	2½	3½	10	8

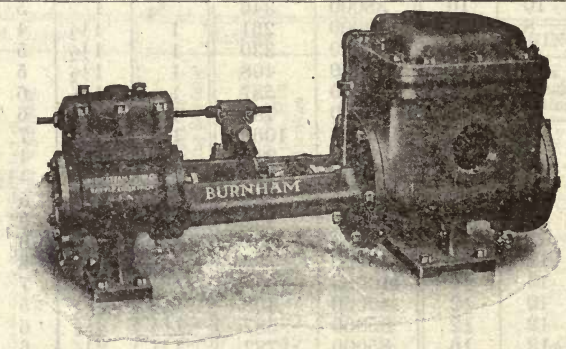


Fig. 89

Burnham Horizontal High Vacuum Pumps

Standard Pattern—Cast-Yoke Pumps

200 Pounds Maximum Steam Pressure, 26 inch Vacuum with 30 inch Barometer.

Size of Pump			Ratings	Size of Openings			
				For long pipe lines use larger pipes, reducing size at the pump openings			
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Displacement in Gals. per Min. at 100 strokes per Min	Steam	Exhaust	Suction	Discharge
4	4	5	27	1½	¾	2	1½
4	5	5	42	1½	¾	2½	2
4	6	5	61	1½	¾	3	2½
4	3½	6	25	1½	¾	2	1½
4	4	6	32	1½	¾	2	1½
4½	4	8	43	1½	¾	2	1½
4½	5	8	68	1½	¾	2½	2
4½	6	8	98	1½	¾	3	2½
4½	7	8	133	1½	¾	3½	3

Burnham Horizontal High Vacuum Pumps (Continued)

Standard Pattern—Cast-Yoke Pumps

200 Pounds Maximum Steam Pressure, 26 inch Vacuum with 30 inch Barometer.

Size of Pump			Ratings	Size of Openings			
				For long pipe lines use larger pipes, reducing size at the pump openings			
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Displacement in Gallons per Min at 100 strokes per Minute	Steam	Exhaust	Suction	Discharge
5	5	6	51	½	¾	2½	2
5	6	6	73	½	¾	3	2½
5	7	6	100	½	¾	3½	3
5	6	10	122	½	¾	3	2½
5	7	10	166	½	¾	3½	3
5	8	10	217	½	¾	4	3½
5	8	12	261	½	¾	4	3½
5½	6	7	85	½	¾	3	2½
5½	7	7	110	½	¾	3½	3
6⅛	6	8	98	¾	1	3	2½
6⅛	7	8	133	¾	1	3½	3
6⅛	8	8	174	¾	1	4	3½
6⅛	7	10	166	¾	1	3½	3
6⅛	8	10	217	¾	1	4	3½
6⅛	9	10	275	¾	1	4½	3½
6⅛	10	10	340	¾	1	5	4
6⅛	8	12	261	¾	1	4	3½
6⅛	10	12	408	¾	1	5	4
7	7	10	166	¾	1	3½	3
7	8	10	217	¾	1	4	3½
7	9	10	275	¾	1	4½	3½
7	10	10	340	¾	1	5	4
8	8	12	261	1	1½	4	3½
8	9	12	330	1	1½	5	4
8	10	12	408	1	1½	5	4
8	12	12	587	1	1½	6	5
8	12	16	782	1	1½	6	5
8	14	16	1065	1	1½	7	6
8½	8	10	217	1	1½	4	3½
8½	9	10	275	1	1½	4½	3½
8½	10	10	340	1	1½	5	4
10	10	12	408	1¼	2	5	4
10	12	12	587	1¼	2	6	5
10	12	16	782	1¼	2	6	5
10	14	16	1065	1¼	2	7	6
12	12	16	782	1½	2½	6	5
12	14	16	1065	1½	2½	7	6

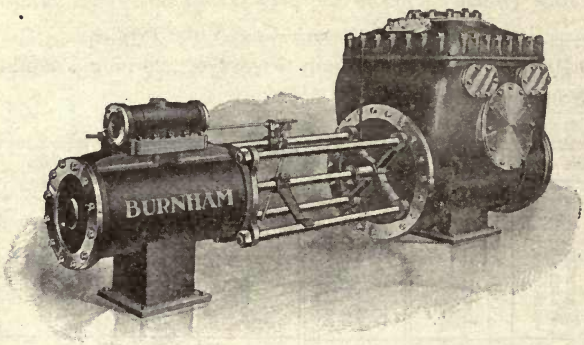


Fig. 155

Burnham Horizontal High-Vacuum Pumps

Standard Pattern—Rod-Yoke Pumps

200 Pounds Maximum Steam Pressure, 26 inch Vacuum with 30 inch Barometer.

Size of Pump			Ratings	Size of Openings			
				For long pipe lines use larger pipes, reducing size at the pump openings			
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Displacement in Gallons per Min. at 100 strokes per Minute	Steam	Exhaust	Suction	Discharge
10	14	20	1330	1½	2½	7	6
10	16	20	1740	1½	2½	8	6
12	14	20	1330	2	2½	7	6
12	16	20	1740	1½	2½	8	6
12	18	20	2200	1½	2½	10	8
14	16	20	1740	2	2½	8	6
14	18	20	2200	2	2½	10	8
14	20	24	3264	2	2½	10	8
16	16	20	1740	2	2½	8	6
16	18	20	2200	2	2½	10	8
16	20	24	3264	2	2½	10	8
18	18	20	2200	2½	3½	10	8
18	20	24	3264	2½	3½	10	8

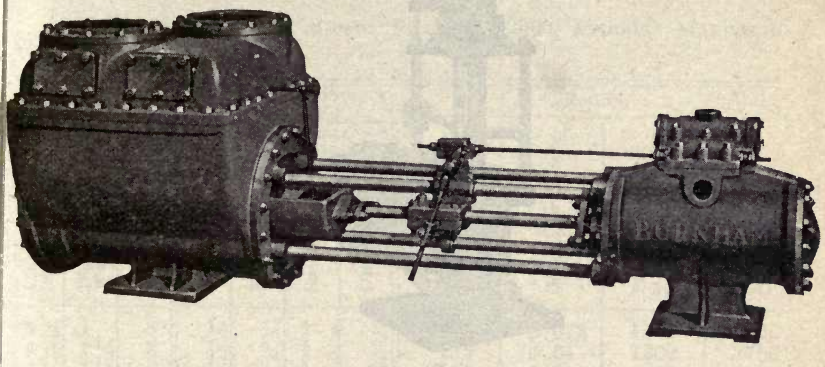


Fig. 137 a

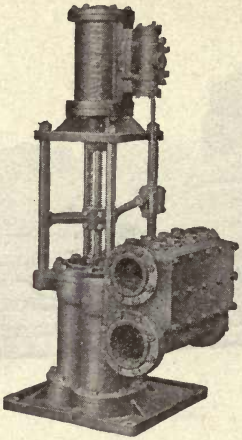
Burnham Horizontal High-Vacuum Pumps

Inverted Suction-Valve Design

200 Pounds Maximum Steam Pressure, 28 inch Vacuum with 30 inch Barometer.

Size of Pump			Ratings	Size of Openings			
				For long pipe lines use larger pipes, reducing size at the pump openings			
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Displacement of G. P. M. at 100 Strokes per Minute	Steam	Exhaust	Suction	Discharge
6⅛	8	10	217	¾	1	5	5
8	10	12	400	1	1½	7	7
8	12	12	587	1	1½	8	8
8	12	16	782	1	1½	8	8
10	14	16	1065	1¼	2	8	8
10	16	16	1392	1¼	2	12	12
10	16	20	1740	1½	2½	12	12
12	18	20	2200	1½	2½	12	12
14	20	24	3264	2	2½	14	14

Burnham Vertical Vacuum Pumps



Burnham Vertical High Vacuum Pumps. 200 Pounds Maximum Steam Pressure, 26 inch Vacuum with 30 inch Barometer.

Fig. 156

Size of Pump			Ratings	Size of Openings For long pipe lines use larger pipes, reducing size at the pump openings.			
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Displacement in Gals. per Minute at 100 strokes per Minute	Steam	Exhaust	Suction	Discharge
4	6	5	61	1/2	3/4	3	2 1/2
4 1/2	6	8	98	1/2	3/4	3	2 1/2
4 1/2	7	8	133	1/2	3/4	3 1/2	3
5	6	6	73	1/2	3/4	3	2 1/2
5	7	6	100	1/2	3/4	3 1/2	3
6 1/8	6	8	98	3/4	1	3	2 1/2
6 1/8	7	8	133	3/4	1	3 1/2	3
6 1/8	8	8	174	3/4	1	4	3 1/2
6 1/8	9	8	220	3/4	1	4	3 1/2
6 1/8	10	8	270	3/4	1	5	4
6 1/8	10	10	340	3/4	1	5	4
6 1/8	12	12	587	3/4	1	6	5
8	10	12	400	1	1 1/2	5	4
8	12	12	587	1	1 1/2	6	5
8	14	12	799	1	1 1/2	7	6
8	16	12	1044	1	1 1/2	8	6

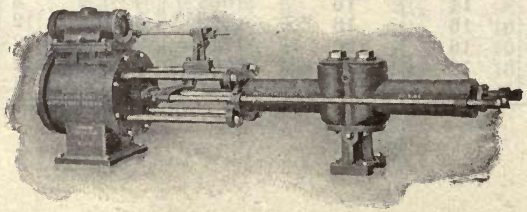


Fig. 110a

Burnham Hydraulic-Pressure Pumps

With Cast-Iron Water Cylinders

200 Pounds Maximum Steam Pressure, 2000 Pounds Maximum Water Pressure.

Size of Pump			Diameter of Pipe Openings				Ratings			
			For long pipe lines use larger pipes, reducing size at the pump openings							
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Strokes per Min	Gallons per Minute	Balanced Pressure at 85 Pounds of Steam	Maximum Pressure for Cast-iron Cylinders
4½	1	8	½	¾	1	¾	135	3.672	1500	2750
4½	1⅛	8	½	¾	1	¾	135	4.64	1200	2750
4½	1¼	8	½	¾	1	¾	135	5.73	990	2750
5½	1	7	½	¾	1	¾	135	3.21	2300	2750
5½	1⅛	7	½	¾	1	¾	135	4.00	1800	2750
5½	1¼	7	½	¾	1	¾	135	5.00	1480	2750
6⅛	1	8	¾	1	1	¾	135	3.672	2870	2750
6⅛	1⅛	8	¾	1	1	¾	135	4.64	2260	2750
6⅛	1¼	8	¾	1	1	¾	135	5.73	1830	2750
6⅛	1¼	10	¾	1	1¼	1	120	6.37	1830	2000
6⅛	1⅜	10	¾	1	1¼	1	120	7.71	1510	2000
6⅛	1½	10	¾	1	1¼	1	120	9.18	1270	2000
6⅛	1⅝	10	¾	1	1¼	1	120	10.77	1087	2000
8½	1¼	10	1	1½	1¼	1	120	6.37	3530	2000
8½	1⅜	10	1	1½	1¼	1	120	7.71	2920	2000
8½	1½	10	1	1½	1¼	1	120	9.18	2450	2000
8½	1⅝	10	1	1½	1¼	1	120	10.77	2090	2000
8	1¼	12	1	1½	1¼	1	100	6.37	3130	2000
8	1½	12	1	1½	1¼	1	100	9.18	2170	2000
8	1¾	12	1	1½	1¼	1	100	12.49	1600	2000
8	2	12	1	1½	1¼	1	100	16.31	1200	2000
8	2½	12	1	1½	1½	1¼	100	25.49	800	2000
10	1½	12	1¼	2	1¼	1	100	9.18	3400	2000
10	1¾	12	1¼	2	1¼	1	100	12.49	2500	2000
10	2	12	1¼	2	1¼	1	100	16.31	1910	2000
10	2½	12	1¼	2	1½	1¼	100	25.49	1200	2000
12	1½	12	1½	2½	1¼	1	100	9.18	4890	2000
12	1¾	12	1½	2½	1¼	1	100	12.49	3590	2000
12	2	12	1½	2½	1¼	1	100	16.31	2750	2000
12	2½	12	1½	2½	1½	1¼	100	25.49	1760	2000
14	1¾	12	2	2½	1¼	1	100	12.49	4900	2000
14	2	12	2	2½	1¼	1	100	16.31	3750	2000
14	2½	12	2	2½	1½	1¼	100	25.49	2400	2000
14	3	12	2	2½	3	2	100	36.71	1660	2000
14	3½	12	2	2½	3	2	100	49.97	1200	2000
16	2½	12	2	2½	1½	1¼	100	25.49	3130	2000
16	3	12	2	2½	3	2	100	36.71	2170	2000
16	3½	12	2	2½	3	2	100	49.97	1590	2000
18	3	20	2½	3½	3	2	60	36.71	2750	2000
18	3½	20	2½	3½	3	2	60	49.97	2020	2000
20	3	20	2½	3½	3	2	60	36.71	3400	2000
20	3½	20	2½	3½	3	2	60	49.97	2500	2000
24	3½	20	3	5	3	2	60	49.97	3600	2000

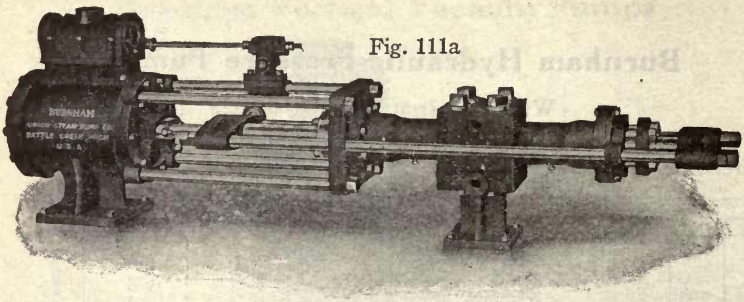


Fig. 111a

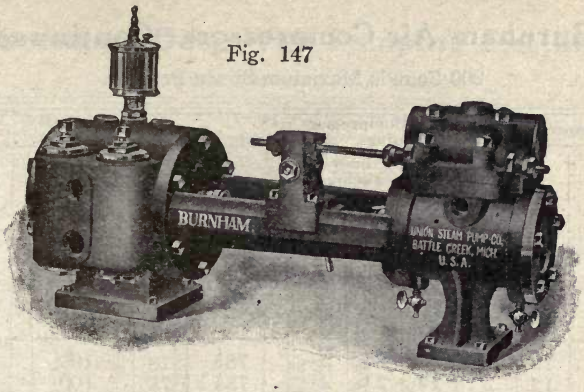
Burnham Hydraulic-Pressure Pumps

With Forged-Steel Water Cylinder

200 Pounds Maximum Steam Pressure, 5000 Pounds Maximum Water Pressure.

Size of Pump			Diameter Pipe Openings				Ratings		
			For long pipe lines use larger pipes, reducing size at the pump openings						
Diameter of Steam Cylinder	Diameter of Water Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Strokes per Minute	Gallons per Min.	Balanced Pressure at 85 Pounds of Steam
4½	⅝	8	½	¾	1	¾	135	2.06	3960
4½	1⅛	8	½	¾	1	¾	135	4.65	1200
5½	¾	7	½	¾	1	¾	135	1.80	4110
5½	1⅛	7	½	¾	1	¾	135	4.05	1800
6⅛	¾	8	¾	1	1	¾	135	2.06	5100
6⅛	1⅛	8	¾	1	1	¾	135	4.65	2260
7	¾	10	¾	1	1	¾	120	2.30	6660
7	1⅛	10	¾	1	1	¾	120	5.05	2960
7	1¼	10	¾	1	1¼	1	120	6.35	2400
8	¾	12	1	1½	1	¾	100	2.30	8650
8	1¼	12	1	1½	1¼	1	100	6.36	3130
8	1½	12	1	1½	1¼	1	100	9.12	2170
8½	¾	10	1	1½	1	¾	120	2.30	9810
8½	1⅛	10	1	1½	1	¾	120	5.05	4370
8½	1¼	10	1	1½	1¼	1	120	6.35	3530
8½	1¾	10	1	1½	1¼	1	120	12.50	1800
10	¾	12	1¼	2	1	¾	100	2.30	13600
10	1⅛	12	1¼	2	1	¾	100	5.16	6050
10	1¼	12	1¼	2	1¼	1	100	6.36	4900
10	1¾	12	1¼	2	1¼	1	100	12.48	2500
12	1¼	12	1½	2½	1¼	1	100	6.36	7050
12	1¾	12	1½	2½	1¼	1	100	12.48	3590
12	2	12	1½	2½	1½	1¼	100	16.32	2750
12	2¾	12	1½	2½	1½	1¼	100	30.84	1450
14	1¼	12	2	2½	1¼	1	100	6.36	9590
14	1¾	12	2	2½	1¼	1	100	12.48	4900
14	2	12	2	2½	1½	1¼	100	16.32	3750
14	2¾	12	2	2½	1½	1¼	100	30.84	1980
16	2	12	2	2½	1½	1¼	100	16.32	4900
16	2¾	12	2	2½	1½	1¼	100	30.84	2590
18	3	20	2½	3½	3	2	60	36.70	2750
18	3¼	20	2½	3½	3	2	60	43.20	2350
20	3	20	2½	3½	3	2	60	36.70	3400
30	4	20	3	5	3	2	60	65.30	4300

Fig. 147



Burnham Air Compressors

200 Pounds Maximum Steam Pressure.

Size of Compressor			Diam. of Compressor Openings				Ratings			
			For long pipe lines use larger pipes, reducing size at the pump openings							
Diameter of Steam Cylinder	Diameter of Air Cylinder	Length of Stroke	Steam	Exhaust	Air Inlet	Air Delivery	Cubic Feet Free Air per Stroke	Cubic Feet Free Air per Minute	Strokes per Minute	Working Air Pressure at 80 lbs. Steam Pressure
4	4	5	½	¾	1	¾	.035	5	160	45 lbs.
4	5	5	½	¾	1	¾	.057	9	160	28 lbs.
4	6⅛	5	½	¾	1½	1½	.085	13	150	19 lbs.
5	4	6	½	¾	1¼	1	.043	6	150	70 lbs.
5	5	6	½	¾	1¼	1	.068	10	150	45 lbs.
5	6⅛	6	½	¾	1½	1½	.098	14	150	30 lbs.
5	7	6	½	¾	1½	1½	.133	20	150	23 lbs.
5½	4	7	½	¾	1¼	1	.05	7	140	85 lbs.
5½	5	7	½	¾	1¼	1	.08	11	140	55 lbs.
5½	6⅛	7	½	¾	1½	1½	.11	15	140	35 lbs.
5½	7	7	½	¾	1½	1½	.165	21	140	27 lbs.
6⅛	4	8	¾	1	1¼	1	.058	7	130	105 lbs.
6⅛	5	8	¾	1	1¼	1	.09	11	130	65 lbs.
6⅛	6⅛	8	¾	1	1½	1½	.13	17	130	45 lbs.
6⅛	7	8	¾	1	1½	1½	.175	22	130	35 lbs.
6⅛	8	10	¾	1	2½	2	.29	32	110	26 lbs.
6⅛	9	10	¾	1	2½	2	.365	40	110	21 lbs.
7	6⅛	10	¾	1	1½	1½	.171	19	110	60 lbs.
7	7	10	¾	1	1½	1½	.223	24	110	45 lbs.
7	8	10	¾	1	2½	2	.29	32	110	34 lbs.
7	9	10	¾	1	2½	2	.365	40	110	27 lbs.
8½	8	10	1	1½	2½	2	.29	32	110	50 lbs.
8½	9	10	1	1½	2½	2	.365	40	110	35 lbs.
8	8	12	1	1½	2½	2	.345	34	100	45 lbs.
8	9	12	1	1½	2½	2	.44	44	100	32 lbs.
8	10	12	1	1½	2½	2½	.545	54	100	26 lbs.
8	12	12	1	1½	2½	2½	.785	78	100	19 lbs.
10	8	12	1¼	2	2½	2	.345	34	100	70 lbs.
10	9	12	1¼	2	2½	2	.44	44	100	55 lbs.
10	10	12	1¼	2	2½	2½	.545	54	100	45 lbs.
10	12	12	1¼	2	2½	2½	.785	78	100	30 lbs.
12	8	12	1½	2½	2½	2	.345	34	100	100 lbs.
12	9	12	1½	2½	2½	2	.44	44	100	80 lbs.
12	10	12	1½	2½	2½	2½	.545	54	100	65 lbs.

Burnham Air Compressors—Continued

200 Pounds Maximum Steam Pressure

Size of Compressor			Diam. of Compressor Openings For long pipe lines use larger pipes, reducing size at the pump openings				Ratings			
Diameter of Steam Cylinder	Diameter of Air Cylinder	Length of Stroke	Steam	Exhaust	Air Inlet	Air Delivery	Cubic Feet Free Air per Stroke	Cubic Feet Free Air per Minute	Strokes per Minute	Working Air Pressure at 80 lbs. Steam Pressure
12	12	12	1½	2½	2½	2½	.785	78	100	45 lbs.
14	12	12	2	2½	2½	2½	.785	78	100	60 lbs.
10	14	16	1¼	2	4½	4	1.42	114	80	22 lbs.
10	16	16	1¼	2	4½	4	1.86	148	80	17 lbs.
12	14	16	1½	2½	4½	4	1.42	114	80	33 lbs.
12	16	16	1½	2½	4½	4	1.86	148	80	24 lbs.
14	14	16	2	2½	4½	4	1.42	114	80	45 lbs.
14	16	16	2	2½	4½	4	1.86	148	80	33 lbs.
16	14	16	2	2½	4½	4	1.42	114	80	60 lbs.
16	16	16	2	2½	4½	4	1.86	148	80	45 lbs.
18	16	16	2½	3½	4½	4	1.86	148	80	55 lbs.
20	16	16	2½	3½	4½	4	1.86	148	80	70 lbs.
14	18	20	2	2½	6	4½	2.94	176	60	27 lbs.
14	20	20	2	2½	6	4½	3.63	218	60	22 lbs.
16	18	20	2	2½	6	4½	2.94	176	60	35 lbs.
16	20	20	2	2½	6	4½	3.63	218	60	29 lbs.
18	18	20	2½	3½	6	4½	2.94	176	60	45 lbs.
18	20	20	2½	3½	6	4½	3.63	218	60	35 lbs.
20	18	20	2½	3½	6	4½	2.94	176	60	55 lbs.
20	20	20	2½	3½	6	4½	3.63	218	60	45 lbs.
16	22	24	2	2½	6	4½	5.27	263	50	24 lbs.
16	24	24	2	2½	6	4½	6.28	314	50	20 lbs.
18	22	24	2½	3½	6	4½	5.27	263	50	30 lbs.
18	24	24	2½	3½	6	4½	6.28	314	50	25 lbs.

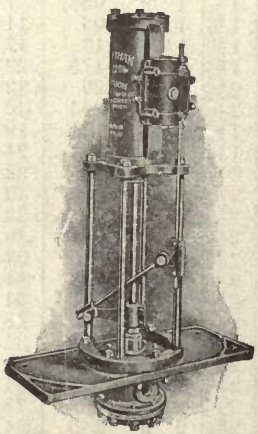


Fig. 157.

Burnham Deep-Well Pumping Engines

150 Pounds Maximum Steam Pressure

Size of Engine		Diameter of Engine openings				
		For Long Pipe Lines use Larger Pipes, Reducing Size at the Engine Openings				
Diameter of Steam Cylinder	Length of Stroke	Steam	Exhaust	Pipe Size of Well Connection	Discharge Pipe	Size of Plunger Rod Connection
5	10	¾	1	2½	1½	¾ pipe
6	16	¾	1	6	2	1⅛
6	24	¾	1	6	2	1⅛
6	36	¾	1	6	2	1⅛
8	24	1	1½	8	3½	1¼
8	36	1	1½	8	3½	1¼
10	24	1½	2	8	3½	1½
10	36	1½	2	8	3½	1½
12	36	1½	2½	8	3½	1½
14	36	2	2½	8	6	1¾
16	42	2	2½	8	6	1¾
18	42	2	3	8	6	2

Smaller openings furnished if desired

Threaded, 8 threads per inch

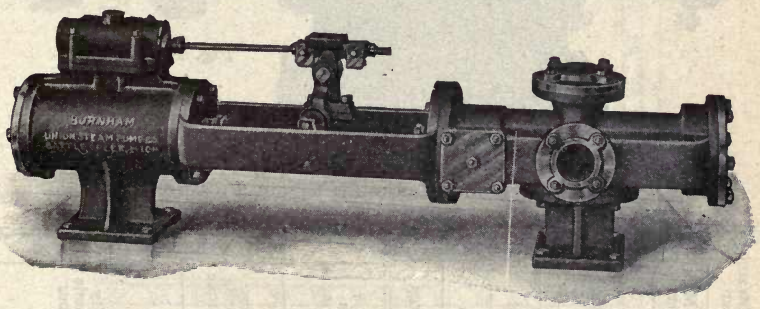


Fig. 146a

Burnham Magma Pumps

200 Pounds Maximum Steam Pressure. 75 Pounds Maximum Liquid Pressure

Size of Pump			Size of Openings				Ratings		
			For long pipe lines use larger pipes, reducing size at pump openings						
Diameter of Steam Cylinder	Diameter of Magma Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Maximum Strokes per Minute	Displacement Gallons per Stroke	*Displacement Gallons per Minute
4	3	5	½	¾	3	2	120	.153	18
6⅛	4	8	¾	1	4	3	75	.434	32
7	5	10	¾	1	5	4	60	849	51
8½	5	10	1	1½	5	4	60	.849	51
8	5	12	1	1½	5	4	50	1.019	51
10	5	12	1¼	2	5	4	50	1.019	51
10	6	12	1¼	2	6	5	50	1.467	73
12	7	12	1½	2½	8	6	50	2.000	100
14	8	12	2	2½	8	8	50	2.610	130
14	10	20	2	2½	10	10	30	6.79	204
16	10	20	2	2½	10	10	30	6.79	204
20	10	20	2½	3½	10	10	30	6.79	204

*The actual amount of liquor handled with a magma pump is 60% of its piston displacement.

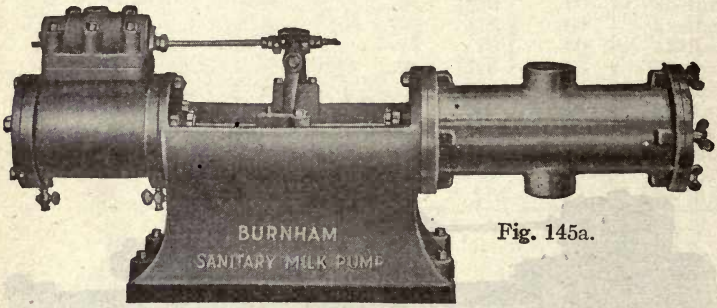


Fig. 145a.

Burnham Sanitary Milk Pumps.

200 Pounds Maximum Steam Pressure

Size of Pump			Size of Openings				Ratings		
Diam. Steam Cylinder	Diam. Milk Cylinder	Stroke	Steam	Exhaust	Suction	Discharge	Strokes Per Minute	Capacity in Pounds Per Stroke	Capacity in Pounds Per Hour
3	3½	3	⅜	½	1½	1½	50–150	1.06	2500 to 9000
4	4	5	½	¾	1½	1½	40–140	2.30	4000 to 19000
5	5	6	½	¾	2	2	30–130	4.32	7500 to 30000
6⅛	7	8	¾	1	3	3	30–120	11.20	20000to 80000

Power Pumps

and

Crank and Fly-wheel Pumps

SECTION SIX

Power Pumps and Crank and Fly Wheel Pumps

Pumps operated through the medium of a crank and connecting rod are classified as Power Pumps, and Crank and Fly Wheel Pumps.

Power Pumps

Power Pumps, as the name indicates are generally operated by the application of power on the crank, which is transmitted through the connecting rod, and crosshead to the water piston or plunger.

Power Pumps are classified with respect to the power and, according to the number of cranks into single, duplex, and triplex.

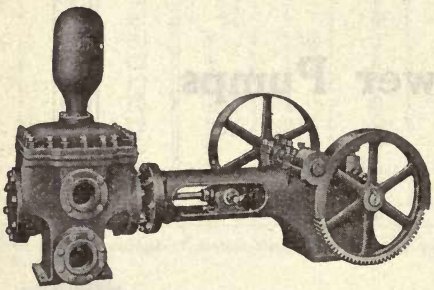
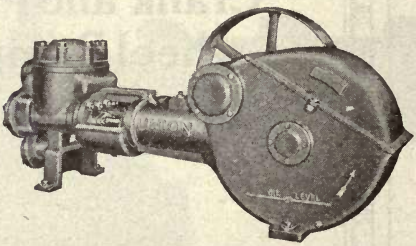


Fig. 158
Single Belt Driven Piston Pump.

Fig. 206.
Duplex Belt-Driven Piston Pump.



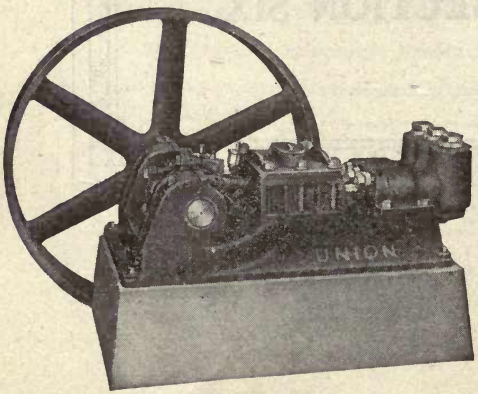


Fig. 160
Triplex Belt Driven Plunger Pump.

Power Pumps are employed in localities where the conditions are such that it is more practical to operate a pump by belt or electricity, than to use a steam driven direct acting pump.

Belt Driven Power Pumps are generally back-geared with one reduction of gearing, the gears have cut teeth, and the pinion is sometimes made of raw hide to reduce the noise.

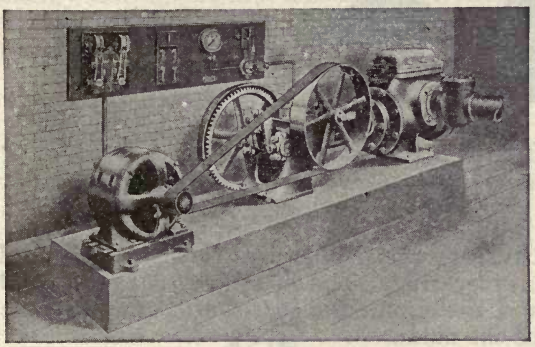


Fig. 161

Plain Belt Driven Vacuum Pump.

Belt Driven Pumps are divided into two types viz. the plain belt drive, and the short belt drive. The former, which is illustrated in figure 161 is operated by its motive power placed at a sufficient distance from same (usually three times the diameter of the driven pulley) to insure a liberal arc of contact on the driving pulley.

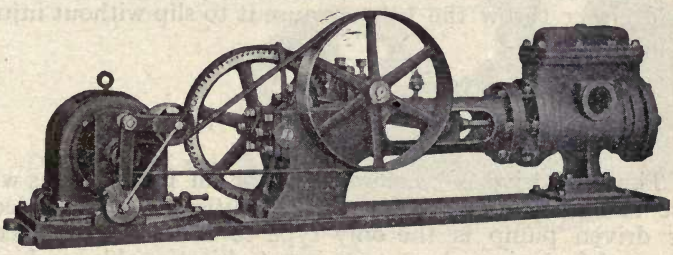


Fig. 162

Belt Driven Vacuum Pump Short Belt Drive.

The short belt drive figure 162 consists of a belt tightener placed on the upper side (the slack side) of the belt, which increases the arc of contact on the driving and driven pulley, and makes it possible to place the pump, and motive power very close together. This type of drive is very compact, and is being used extensively.

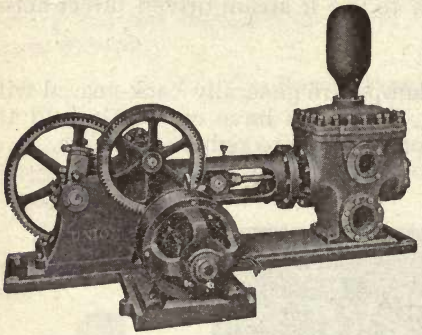


Fig. 163
Electric Gear Driven
Piston Pump

Electric gear driven power pumps are usually backgeared with two reductions of gearing. The gears have cut teeth, and the motor pinion, and sometimes the gear-shaft pinion is made of raw hide to reduce the noise.

The type of drive to employ is purely a problem for the user to solve, as the local conditions are the determining factor. The belt drive is the most practical type of drive for the reasons that it operates with a minimum amount of noise, and if a suddenly applied shock or excess pressure is put on the water end, it will either throw the belt or cause it to slip without injuring the pump.

The belt drive is recommended for all installations where noise is obectionable. There are localities where the electric gear driven pump is the only type to use. For instance, in mines and facturies, where noise is not objectionable, and where on account of the moisture, etc., it is not practical to use the belt drive.

The chief advantage of the electric gear drive is that it is very compact.

Crank and Flywheel Pumps.

Crank and flywheel pumps are generally steam driven, and the power is applied through the steam piston and piston rod to the water piston or plunger, the reciprocating action of which is transmitted through the crosshead, and connecting rod to the crank. In this type of pump the fly wheels store up the energy during the first half of the stroke, in order to replenish it during the remainder of the stroke, with the result that the steam may be used expansively.

Crank and flywheel pumps are classified with respect to the power end, according to the number of cranks into single and duplex pumps.

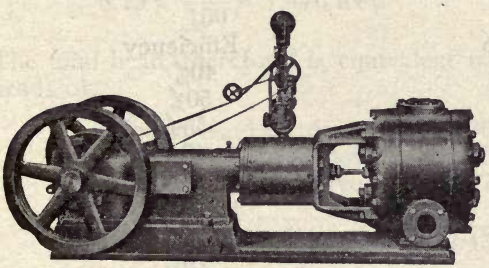


Fig. 91

Single Crank and Flywheel Dry Vacuum Pump.

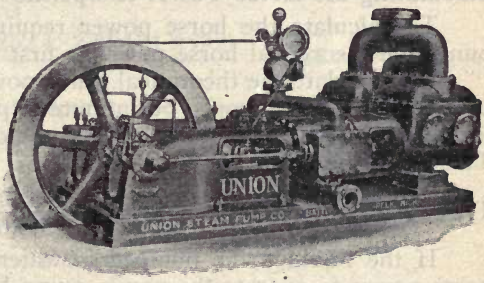


Fig. 164

Duplex Crank and Flywheel Wet Vacuum Pump

The crank and flywheel pump, on account of being able to use the steam extensively, is used where economy is of prime importance.

Power pumps and crank and flywheel pumps are classified with respect to the type of water end into piston and plunger pumps.

Water End Classification

The different types of water ends employed on power and crank and fly wheel pumps shown herein, the manner of fitting same for different classes of service, and in fact all information respecting the water end of the pump is given in Section Five.

Efficiency

In calculating the horse power required to operate power pumps it is necessary to take into consideration the efficiency of the pump.

The efficiency E is the ratio of the water horse power to the brake horse power. That is

$$E = \frac{\text{Water Horse Power}}{\text{Brake Horse Power}}$$

The efficiency of power pumps varies with conditions of operation, and this factor is obtained only from actual test.

The following table gives the mechanical efficiency of power pumps, and for calculating purposes it is advisable to multiply these figures by 90% in order to take care of any unforeseen losses.

Size of Pump	Efficiency
4" stroke	40%
5" "	50%
6" "	55%
8" "	60%
10" "	65%
12" "	70%
16" "	72%
20" "	75%

Calculating the Horse Power to Operate Power Pressure Pumps

To calculate the horse power required to operate a power pump the theoretical horse power is first determined.

To calculate the theoretical horse power multiply the weight of the liquid to be pumped per minute in pounds by the total head in feet that it is to be pumped, and divide this result by 33000. This gives the following formula:

$$\text{Theo. H. P.} = \frac{\text{Wt. liquid per minute in pounds} \times \text{head in feet}}{33{,}000}$$

If the liquid to be pumped is water, then the formula for horse power to operate the power pump becomes

$$\text{Horse Power} = \frac{W \times H}{33{,}000 \times E}$$

Where W=Weight of liquid pumped per minute in pounds.
H=The total head in feet.
E=The efficiency of the pump.

For a pump handling water

$$\text{H. P.} = \frac{.000252 \times H \times G}{E} \qquad (61)$$

Where H=The total heat in feet.
G=Gallons per minute.
E=The efficiency of the pump.

Example

It is desired to elevate 300 gallons per minute of water to an elevation of 200 feet through 300 lineal feet of 4" pipe with three 90° elbows, and one 4" globe valve. What horse power is required to operate a power pump for this duty, assuming that it has an efficiency of 70%?

The friction loss of 300 gallons per minute through 300 feet of 4" pipe from the table on page 145 is equal to 18.30 feet. The friction loss of 300 gallons per minute through three 90° elbows from the table on page 147 is equal to 2.658 feet. The friction loss of 300 gallons per minute through one 4" globe valve from the table on page 148 is equivalent to 30 feet of 4" pipe, and from the table on page 145 the friction loss of 300 gallons per minute through 30 feet of 4" pipe=

$$6.15 \times \frac{24}{100} = 1.476 \text{ feet}$$

The total head, therefore, is equivalent to the sum of the above heads or,

Friction head in pipe	=	18.30'
" " " elbow	=	2.658'
" " " valve	=	1.476'
Static Head	=	200'
Total Head		222.434'

Then the horse power to operate the pump from formula 61 assuming 70% efficiency is

$$\text{H. P.} = \frac{.000252 \times 222.434 \times 300}{70} = 24$$

In selecting a motor to operate the power pump it would be advisable to use a 25 horse power motor.

Horse Power to Operate Power Wet Vacuum Pumps

To calculate the horse power to operate a power wet vacuum pump, formula 61 may be used by substitnting for H 35 feet (which is the equivalent of 15 lbs. pressure), and for G the displacement of the pump in gallons per minute. The mechanical efficiency given on page 384 is applicable to this type of pump.

Having determined the horse power required, the cost of operating the pump per hour can be obtained by multiplying this figure by the cost per horse power hour of operating the motor or engine used to drive the pump.

Types of Motors to Use With Power Pumps

For power pumps we recommend for direct current motors the Compound Wound type, and in alternating currents, the Slip Ring type of motor.

The types of controls to use for the motors are fully given in Section 4.

Data Required for Estimates for Power Pumps and Crank and Fly Wheel Pumps.

1. For what purpose is pump to be used?
2. (a) Capacity of pump in U. S. gallons per minute.
   (b) If the pump is for vacuum service, give the number of square feet of radiation, or the number of cubic foot displacement per minute required.
   (c) If pump is for use with a condenser, give the number of pounds of steam to be condensed per hour, the temperature of condensing water, the vacuum to be carried and the type of condenser.
   (d) If pump is for evaporator service give the nature of the liquor to be evaporated, the quantity of liquor to be evaporated per hour, the temperature of condensing water, the vacuum under which the liquid is to be evaporated, and the number of effects in the evaporator?
3. Total lift, including suction discharge lift, and pipe friction in feet.
4. Length and diameter of suction pipe?
5. Vertical distance from water level to pump in feet?
6. Number and size of elbows in suction pipe?
7. Length and diameter of discharge pipe?
8. Vertical distance above pump, or against what pressure is liquid to be discharged?
9. Number and size of elbows in discharge pipe?
10. Number and diameter of valves in discharge pipe?
11. Temperature of liquid in degrees Fah.?
12. Specific gravity of liquid?
13. Nature of liquid to be handled: Fresh water, salt water, acidulous, alkaline, gritty, etc.?
14. If pump is to be motor driven, state characteristics of current? If direct current, give voltage, if alternating current give voltage, cycles, and phase?
15. If pump is to be belt driven give the dimensions and speed of the driving pulley?
16. If pump is to be of the crank and flywheel type, give the lowest steam pressure to be used at the pump?
17. (a) Will pump exhaust into the atmosphere?
    (b) Will pump exhaust into a heater (State whether open or closed).
18. What pressure will pump exhaust against?
19. If pump is to operate condensing give vacuum to be carried on the condenser.
20. Where is pump to be located?

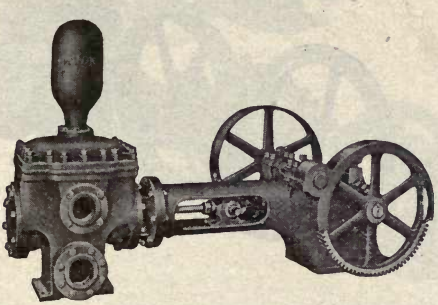


Fig. 158
Single Belt Driven
Piston Pump.

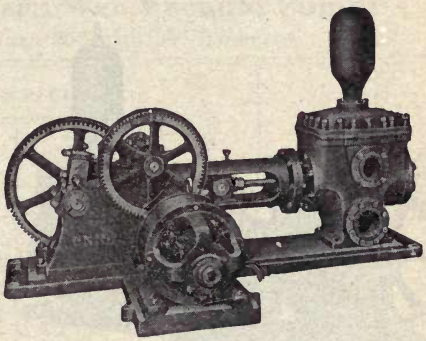


Fig. 163
Electric Gear Driven
Piston Pump

Single Piston Pattern Pressure Pumps

150 Pounds Maximum Pressure

SIZE	Suction, Inches	Discharge Inches	Gallons per Revolution	Maximum Revolutions Crank Shaft per Minute	Gallons per Minute	Gallons per Hour	Size of Pulley	Gear Ratio
2½x 5	1½	1¼	.20	75	15	900	16x3	4.83
3 x 6	2	1½	.36	75	27	1600	16x3	4.83
3½x 6	2½	2	.50	75	37	2220	16x3	4.83
4 x 8	2½	2	.87	60	52	3120	20x5	5.4
4½x 8	3	2½	1.10	60	66	3960	20x5	5.4
5 x12	3½	3	2.00	40	80	4800	32x6	5.12
6 x12	4	3	3.00	40	120	7200	32x6	5.12
6½x12	4	3	3.45	40	138	8280	32x6	5.12
7 x12	5	4	4.00	40	160	9600	36x8	5
7½x12	5	4	4.60	40	184	11000	36x8	5
8 x12	5	4	5.22	40	209	12540	36x8	5
9 x12	6	5	6.60	40	264	15840	42x8	5
10 x12	8	6	8.15	40	326	19560	42x8	5

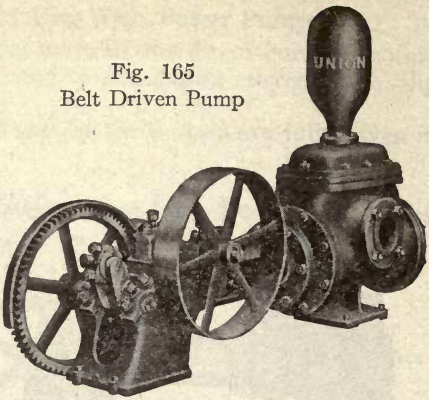


Fig. 165
Belt Driven Pump

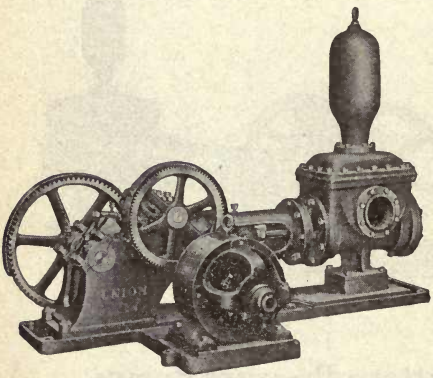


Fig. 166
Motor Driven Pump

Single Piston Pattern Light Service Pumps

75 Pounds Maximum Pressure

SIZE	Maximum Pressure in Pounds	Suction Inches	Discharge, Inches	Gallons per Revolution	Maximum Rev. Crank Shaft Per Minute	Gallons per Minute	Gallons per Hour	Size of Pulley	Gear Ratio
4 x 6	75	2½	2	.65	75	49	2940	16x3	4.83
4½x 6	60	3	2½	.83	75	61.5	3690	16x3	4 83
5 x 8	75	3½	3	1.36	60	81.5	4890	20x5	5.4
5½x 8	75	3½	3	1.65	60	99	5922	20x5	5.4
6 x 8	75	4	3½	1.96	60	117	7020	20x5	5.4
7 x 8	60	5	4	2.66	60	160	9600	20x5	5.4
8 x 8	50	5	4	3.48	60	209	12540	20x5	5.4
8 x12	75	5	4	5.22	40	209	12540	32x6	5.12
9 x12	60	6	5	6.61	40	264	15800	32x6	5.12
10 x12	50	8	6	8.16	40	326	19560	32x6	5
10 x12	75	8	6	8.16	40	326	19560	32x6	5.12
12 x12	75	8	6	11.74	40	469	28140	36x8	5
14 x12	75	10	8	16	40	640	38400	42x8	5
16 x12	60	10	8	20.88	40	835	50100	42x8	5

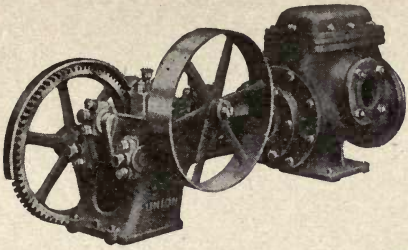


Fig. 167
Belt Driven Pump

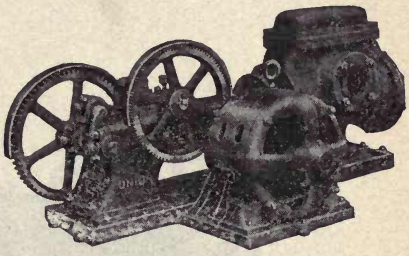


Fig. 168
Motor Driven Pump

Standard Pattern Wet Vacuum Pumps

Maximum Vacuum 26″ with 30″ Barometer

Size of Pump	Suction	Discharge	CAPACITY			Size of Pulley	Ratio of Gears	Horse Power of Motor Required
			Square Feet of Radiating Surface	Rev. per Minute Crankshaft	Gals. Displacement per Min.			
3½x 5	2	1½	2000	40	17	12x2½	4¾ to 1	¾
4 x 5	2	1½	2600	40	21	12x2½	4¾ to 1	¾
5 x 5	2½	2	4000	40	34	16x3	4¾ to 1	1
6 x 5	3	2½	5000	40	49	16x3	4¾ to 1	2
5 x 6	2½	2	5000	40	40	16x3	4¾ to 1	2
6 x 6	3	2½	7000	40	59	16x3	4¾ to 1	2
7 x 6	3½	3	9000	40	80	16x3	4¾ to 1	2
8 x 6	4	3½	12000	40	104	16x3	4¾ to 1	3
5 x 8	2½	2	6500	40	54	18x3½	5½ to 1	2
6 x 8	3	2½	9000	40	78	18x3½	5½ to 1	2
7 x 8	3½	3	12500	40	107	18x3½	5½ to 1	3
8 x 8	4	3½	16000	40	139	18x3½	5½ to 1	3
9 x 8	4½	3½	20000	40	176	18x3½	5½ to 1	5
10 x 8	5	4	24000	40	228	18x3½	5½ to 1	5
6 x10	3	2½	11000	40	98	20x5	5½ to 1	3
7 x10	3½	3	14800	40	133	20x5	5½ to 1	3
8 x10	4	3½	19500	40	175	20x5	5½ to 1	5
9 x10	4½	3½	24500	40	220	20x5	5½ to 1	5
10 x10	5	4	30000	40	270	20x5	5½ to 1	5
12 x10	6	5	40000	40	390	20x5	5½ to 1	7½
8 x12	4	3½	23000	40	208	20x5	6 to 1	5
9 x12	5	4	30000	40	263	24x5	6 to 1	5
10 x12	5	4	37000	40	325	24x5	6 to 1	5
12 x12	6	5	50000	40	470	24x5	6 to 1	7½
14 x12	7	6	70000	40	640	24x5	6 to 1	10
12 x16	6	5	50000	30	470	24x5	5 to 1	7½
14 x16	7	6	70000	30	640	28x5	5 to 1	10
16 x16	8	6	93000	30	835	28x5	5 to 1	10
18 x16	10	8	117000	30	1050	32x6	5 to 1	15
14 x20	7	6	70000	25	665	28x5	4¾ to 1	10
16 x20	8	6	93000	25	870	28x5	4¾ to 1	12
18 x20	10	8	117000	25	1100	32x6	4¾ to 1	15
20 x20	10	8	145000	25	1360	32x6	4¾ to 1	20

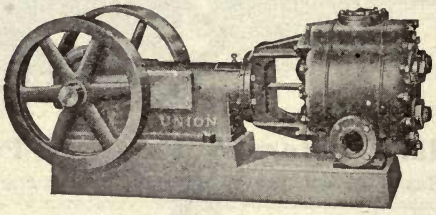


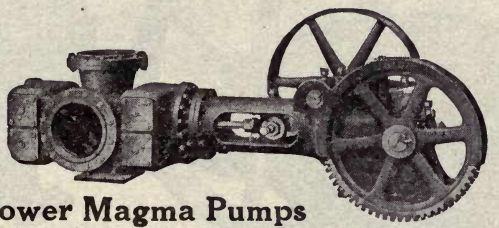
Fig. 93

Single Belt-Driven Dry-Vacuum Pumps

28½"-29" Vacuum, 30" Barometer

Air Cylinder		Revolutions per Minute	Displacement Cubic Feet Free Air		Brake Horse Power Required at Peak Load	Pipe Openings			Driving Pulley		Diameter of Fly-Wheel
Diameter	Stroke		Per Revolution	Per Minute		Suction	Discharge	Cooling Water Inlet	Diameter	Width of Belt	
10	6	275	.545	150	6	3	2½	½	28	6	28
14	6	275	1.06	292	12	4	3½	½	28	6	28
18	6	275	1.76	483	18	5	5	½	28	6	28
18	8	250	2.35	590	22	5	5	½	42	8	42
22	8	250	3.52	880	31	7	6	½	42	8	42
22	10	235	4.40	1035	37	7	6	½	48	10	48
26	10	235	6.15	1445	49	8	7	½	48	10	48
28	12	220	8.56	1880	65	9	8	½	66	12	55
30	12	220	9.82	2160	75	10	8	½	66	12	55
32	15	210	13.98	2940	100	12	10	½	72	15	60

Fig. 170



Power Magma Pumps

75 Pounds Maximum Pressure

Size of Pump		Size of Openings		Displacement of Piston				
Diameter of Magma Cylinder	Stroke	For long pipe lines, use larger pipes, reducing size at pump openings: Suction	Discharge	Gallons per Revolution	R. P. M.	Gallons per Minute*	Size of Pulley Belt-Driven Pump	Gear Ratio Belt-Driven Pump
3	5	3	2	.31	50	15	16 x 3	4¾
4	8	4	3	.88	35	30	20 x 5	5½
5	10	5	4	1.70	30	51	20 x 5	5
6	12	6	5	2.92	25	73	24 x 5	5
7	12	8	6	4.00	25	100	24 x 5	5
8	12	8	8	5.22	25	130	32 x 6	5
10	12	10	10	8.16	25	200	32 x 6	5
12	12	12	12	11.74	25	293	36 x 8	5

*The actual amount of liquor handled with a magma pump is 60% of its piston displacement.

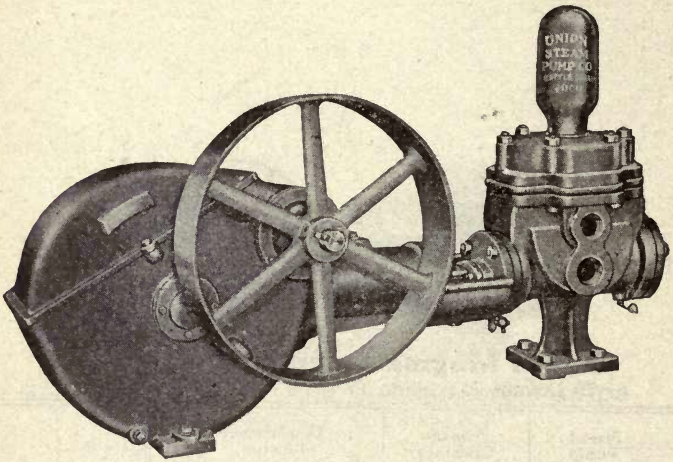


Fig. 207.

Single Power Pumps

Self-Oiling Type

231 Feet Maximum Pressure.

SIZE OF PUMP	Speed, R. P. M.	Gallons per Minute	Gallons per Hour	H. P. Recommended for 231 ft. lift	Pulley Belt	Suction Pipe	Discharge Pipe
3 x 4	60	14.5	870	1¾	14 x 2½	2	1½
4 x 5	50	27.0	1620	3	16 x 4	2½	2
5 x 5	50	42.5	2550	4½	16 x 4	2½	2½
6 x 6	50	73.	4380	8.0	24 x 4	3	3

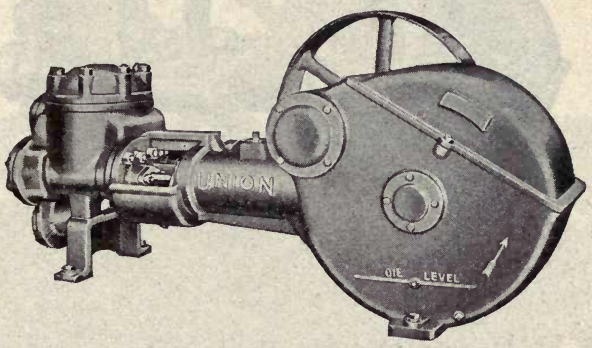


Fig. 206.

Belt-Driven Pump.

Duplex Power Pumps

Self-Oiling Type

Size		Maximum Water Pressure in Pounds	Max. Revolutions of Crankshaft per Minute	Capacity		Pipe Openings		Pulleys			Gear Ratio
									Belt		
Dia. Water Cylinders	Stroke			Gals. per Revolution	Gals. per Minute	Suction	Discharge	Diameter	Width	Single or Double	
2½	4	250	75	.34	25	2	1¼	16	3	S	
2¾	4	200	75	.41	30	2	1½	16	3	S	
3	4	150	75	.49	36	2½	1½	16	3	S	
3¾	4	125	75	.76	57	2½	1½	16	3	S	
4	4	100	75	.87	66	2½	1½	18	3½	S	
4½	4	75	75	1.10	82	3	2	18	3½	S	
4¾	4	60	75	1.22	92	3	2	18	3½	S	
3	6	300	60	.72	43	3	2	24	5	D	
3½	6	250	60	.90	59	3	2	24	5	D	
4	6	200	60	1.32	79	3	2	24	5	D	
4¾	6	140	60	1.84	110	4	3	24	5	D	
5	6	125	60	2.04	122	4	3	24	5	D	
5½	6	100	60	2.46	148	4	3	24	5	D	
5¾	6	95	60	2.69	161	4	3	24	5	D	
6	6	85	60	2.93	175	4	3	24	5	D	
4½	10	300	50	2.75	138	4	3	36	8	D	
5¼	10	250	50	3.74	187	4	3	36	8	D	
6	10	200	50	4.89	244	5	4	36	8	D	
7	10	150	50	6.66	333	6	5	36	8	D	
8½	10	100	50	9.80	490	6	5	36	8	D	
10	10	75	50	13.6	680	8	6	36	8	D	

5:1

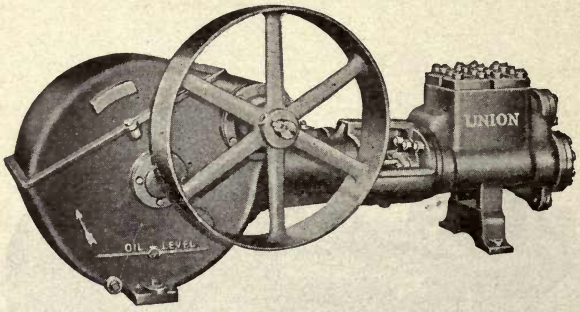


Fig. 208.

Union Duplex Power Pumps

Oil Line Pumps

Self-Oiling Type

Size		Maximum Oil Pressure	Max. Revolutions of Crankshaft per Minute	Capacity			Openings Pipe		Pulleys			Gear Ratio
										Belt		
Dia. Oil Cylinders	Stroke			Gals. per Revolution	Gals. per Minute	Barrels per Hour (42 Gallons per Barrel)	Suction	Discharge	Diameter	Width	Single or Double	
3	6	350	60	.74	44	63	2½	2	24	5	D	5:1
3½	6	250	60	.99	59	84	2½	2	24	5	D	
3	8	450	55	.98	54	77	2½	2	28	6	D	
3½	8	350	55	1.33	73	104	2½	2	28	6	D	
3	10	800	50	1.22	61	87	3	2	36	8	D	
3½	10	600	50	1.67	84	120	3	2	36	8	D	
4	10	450	50	2.17	108	154	4	3	36	8	D	
4½	10	350	50	2.75	137	196	4	3	36	8	D	

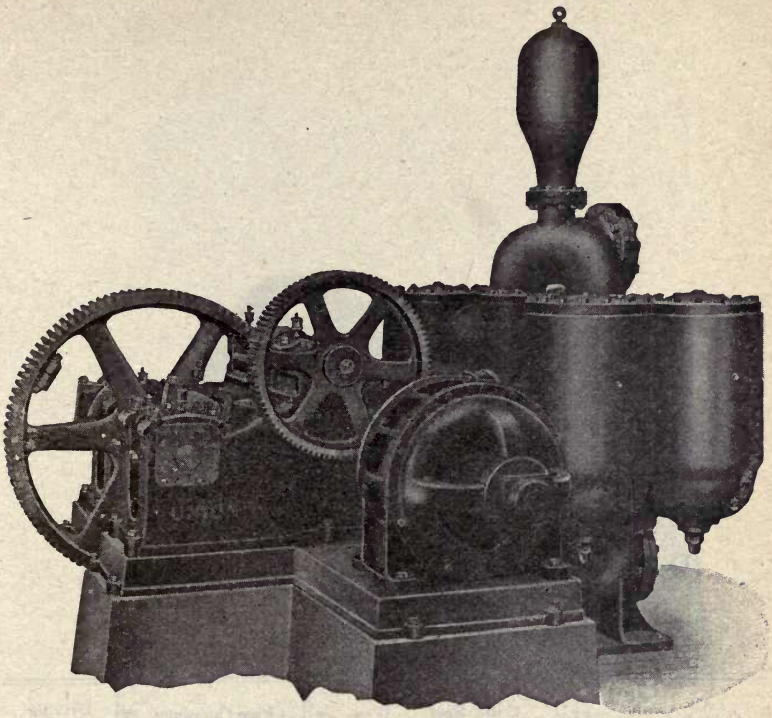
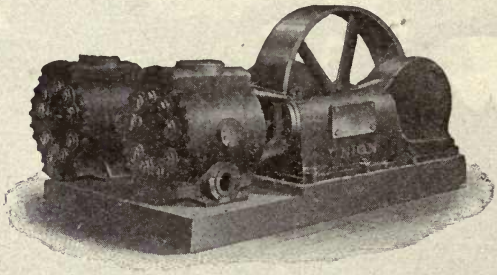


Fig. 185

Union Duplex Power Cargo Oil Pumps

Size of Pump		Diameter of Pump Openings		RATINGS				
Diameter Oil Cylinders	Length Stroke	Suction	Discharge	Max. Pressure Lbs. Sq. Inch	R. P. M.	Gallons per Revolution	Gallons per Minute	Barrels (42 Gallons) per Hour
10	12	10	8	125	40	16.3	652	931
10½	12	10	8	110	40	17.8	712	1020
11	12	10	8	100	40	19.7	788	1120
12	15	14	12	125	40	29.4	1176	1670
13	15	14	12	100	40	34.5	1380	1970
14	15	14	12	90	40	40.0	1600	2280

Fig. 173

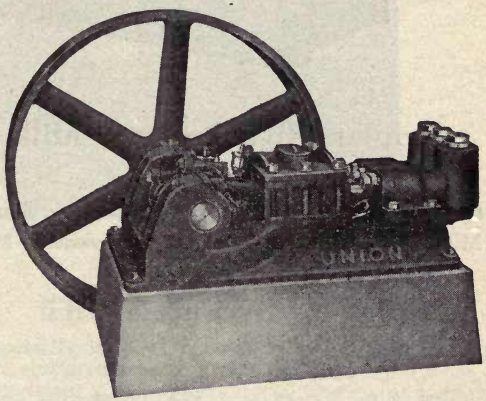


Duplex Belt Driven Enclosed Type Dry Vacuum Pumps

28½"-29" Vacuum 30" Barometer

Air Cylinders			Displacement Cubic Feet Free Air			Pipe Openings			Driving Pulley	
Diameter	Stroke	Revolutions per Minute	Per Revolution	Per Minute	Brake H. P. Required at Peak Intake Atm. Discharge	Suction Each	Discharge Each	Cooling Water Inlet	Diameter	Width of Belt
18	8	250	4.70	1180	44	5	5	¾	54	10
22	8	250	7.04	1760	62	7	6	¾	54	10
22	10	235	8.80	2070	74	7	6	1	54	14
26	10	235	12.30	2890	98	8	7	1	54	14
28	12	220	17.12	3760	130	9	8	1	66	18
30	12	220	19.64	4320	150	10	8	1	66	18
32	15	210	27.96	5880	200	12	10	1	80	25

Fig. 160

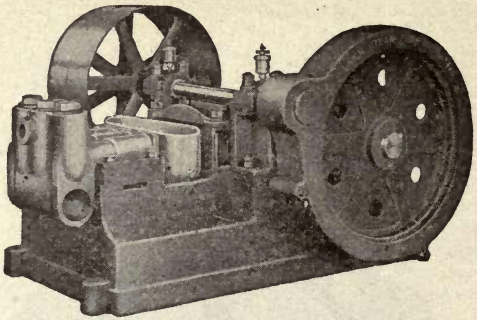


Triplex Belt Driven Plunger Pumps

2000 Pounds Maximum Pressure

SIZE	Speed R. P. M.	Capacity in Gals. per Minute	Size of Openings		Size of Pulley
			Suction	Discharge	
1 x 3	150	4½	1	½	24 x 5
1¼ x 4	150	9½	1½	1	48 x 6
1½ x 4	150	13½	1½	1	48 x 6

Fig. 174



Triplex High Pressure Milk Pumps

A special type of Triplex Power Pump for Spraying or Atomizing Condensed Milk in the manufacture of Dry or Powdered Milk.

Table of Capacities, Pressures, etc.

Size		Max. R. P. M.	Pounds per Revolution	Pounds per Hour	Maximum Pressure	Pulley
Plunger	Stroke					
1	3	100	.275	1650	3000	16 x 3
1	4	100	.367	2200	3500	20 x 5
1¼	4	100	.572	3430	3000	20 x 5
1½	4	100	.820	4920	2000	20 x 5
1¾	4	60	1.10	4607	900	20 x 5
2	4	60	1.47	5286	900	20 x 5

NOTE.— Capacities are based on milk weighing nine pounds per gallon after condensing 5 to 1.

Suction opening on 3-inch stroke pumps, 1¼-inch standard pipe tap. 1½-inch sanitary union.

Discharge opening on 3-inch stroke pumps, ½-inch standard pipe tap.

Suction opening on 4-inch stroke pumps, 1½-inch standard pipe tap.

Discharge opening on 4-inch stroke pumps, 1-inch standard pipe tap.

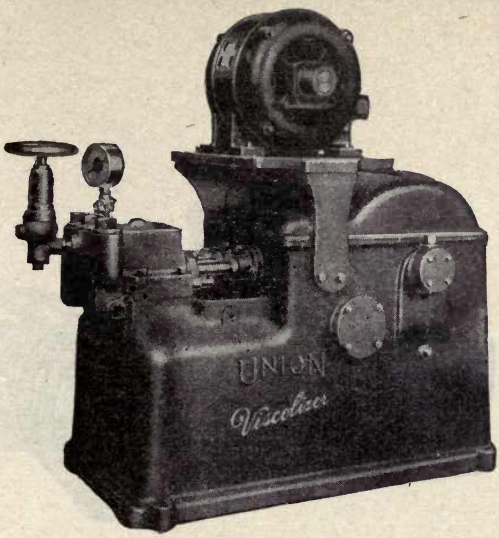


Fig. 209.

The Viscolizer

The Viscolizer is a specially designed, powerfully constructed triplex pump used in the manufacture and processing of liquid and semi-liquid foods, medicine, drugs and oil emulsions. The possibility of storing and marketing evaporated milk is due entirely to the ability of this machine to prevent separation of the butter fat from the milk.

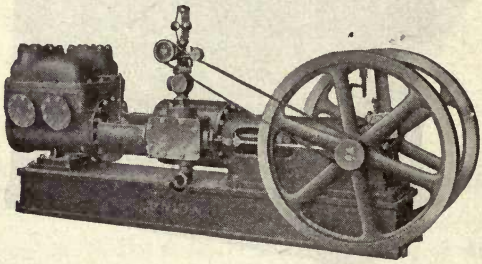
In salad dressings and medicines or other oil emulsions of similar character the oil is so thoroughly mixed with the water by being forced under high pressures through the specially constructed emulsifying device that it will not separate and can therefore be marketed to advantage.

The process is relatively new and requires primarily a sturdy pump which will maintain even pressures in the emulsifying device. These requisites are best met by the triplex design and especially by that shown in figure 209.

Sizes, Capacities and Installation Data

Size	Gallons per Hour	Drive Pulley	Speed R. P. M. Pulley	Gear Ratio	Crank Speed	Motor H. P.	Shipping Weight	Floor Space Required	Intake Sanitary Piping	Outlet Sanitary Piping
100	100	12x2½	480	4 to 1	120	3	900	30x36	1	1
200	200	16x3	500	5 to 1	100	7½	1500	30x48	1½	1½
300	300	20x5	500 to 600	6 to 1	100	10	2300	48x60	1½	1½
450	450	20x5	500 to 600	6 to 1	90 to 100	15	2500	50x66	1½	1½
800	800	28x6	450	5 to 1	90	25	4000	56x78	2	1½

Fig. 176



Single Crank and Flywheel Wet Vacuum Pumps

28″ Vacuum, with 30″ Barometer

DESIGN	Size of Pump		Diameter of Pump Openings						Displacement in Gallons per Minute, at 50 Revolutions per Minute
	Diameter of Steam Cylinder	Diameter of Pump Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Charging Pipes (when ordered)	
Center Crank	6	8	6	1	1½	5	5	¾	130
	8	9	8	1½	2	6	6	¾	220
	8	9	8	1½	2	6	6	¾	220
	8	10	8	1½	2	7	7	1	272
	8	12	8	1½	3	7	7	1¼	390
	10	12	10	2	3	7	7	1¼	490
	10	14	10	2	3	8	8	1¼	665
	10	14	12	2	3	8	8	1¼	800
	12	14	12	2½	3½	8	8	1¼	800
	12	16	12	2½	5½	12	12	1½	1040
	12	18	12	2½	3½	12	12	1½	1320
Side Crank	10	16	20	1½	2	12	12	1½	1740
	12	16	20	2	3½	12	12	1½	1740
	12	18	20	2	3½	12	12	1½	2200
	14	16	20	2½	4	12	12	1½	1740
	14	18	20	2½	4	12	12	1½	2200
	16	18	20	3½	4½	12	12	1½	2200

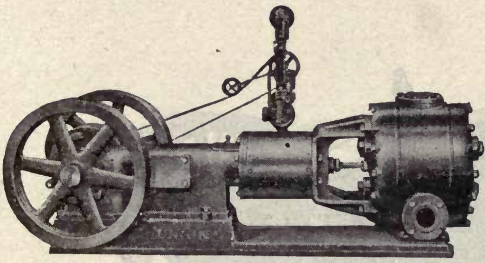


Fig. 91

Single Crank and Flywheel Dry Vacuum Pumps

28½"-29" Vacuum 30" Barometer

SIZE OF PUMP				Displacement Cu. Ft. Free Air				PIPE OPENINGS					
Diameter Steam Cylinder	Diameter Air Cylinder	Stroke	Revolutions per Minute	Per Revolution	Per Minute	Required Steam Pressure	I.H.P. of Steam Cylinder at Peak Intake Atmospheric Discharge	Steam	Exhaust	Suction	Discharge	Cooling Water Inlet	Diameter of Fly-Wheel
6⅛	10	6	275	.545	150	50	6	1½	2	3	2½	½	28
6⅛	14	6	275	1.06	292	60	12	1½	2	4	3½	½	28
6⅛	18	6	275	1.76	483	100	18	1½	2	5	5	½	28
8	14	6	275	1.06	292	35	12	2	2½	4	3½	½	28
8	18	6	275	1.76	483	50	18	2	2½	5	5	½	28
8	18	8	250	2.35	590	50	22	2	2½	5	5	½	36
8	22	8	250	3.52	874	90	31	2	2½	7	6	½	36
10	18	8	250	2.35	590	35	22	2½	3	5	5	½	36
10	22	8	250	3.52	874	50	31	2½	3	7	6	½	36
10	22	10	235	4.40	1035	70	37	3	3½	7	6	½	48
10	26	10	235	6.15	1432	85	49	3	3½	8	7	½	48
12	22	10	235	4.40	1035	35	37	3½	4	7	6	½	48
12	26	10	235	6.15	1432	50	49	3½	4	8	7	½	48
12	28	12	220	8.56	1880	80	65	3½	4	9	8	½	55
12	30	12	220	9.82	2160	100	75	3½	4	10	8	½	55
14	28	12	220	8.56	1880	40	65	4	5	9	8	½	55
14	30	12	220	9.82	2160	45	75	4	5	10	8	½	55
14	32	15	210	13.98	2940	80	100	4	5	12	10	½	60
18	32	15	210	13.98	2940	35	100	5	6	12	10	½	60

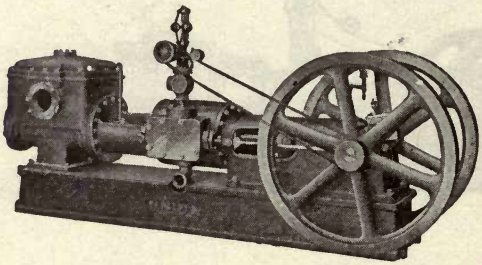
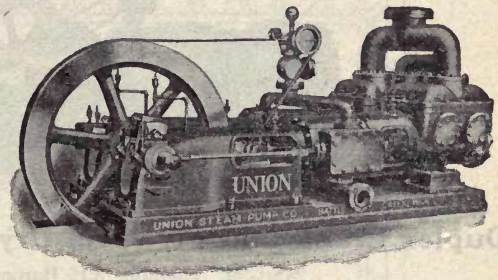


Fig. 177

Single Crank and Flywheel Syrup Pumps

Size of Pump			Dia. of Pump Openings				Displacement in Gallons per Minute, at 50 Revolutions per minute
			For long pipe lines use larger pipes, reducing size at pump openings				
Diameter of Steam Cylinder	Diameter of Pump Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	
6	4	6	1	1½	2½	2	33
6	5	6	1	1½	3½	3	61
6	6	6	1	1½	4	3½	7
6	7	6	1	1½	5	4	100
6	8	6	1	1½	5	4	130
8	5	8	1½	2	3½	3	82
8	6	8	1½	2	4	3½	97
8	8	8	1½	2	5	4	175
8	10	8	1½	2	6	5	272
10	6	10	2	3	4	3½	121
10	8	10	2	3	5	4	218
10	10	10	2	3	6	5	340
10	12	10	2	3	8	6	485
12	10	12	2½	3	8	6	410
12	12	12	2½	3	8	6	585

Fig. 164



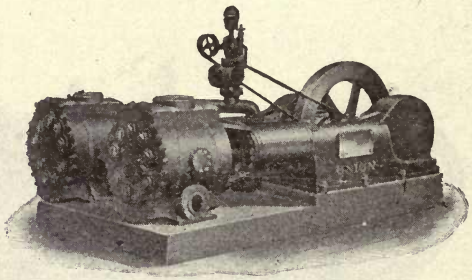
Duplex Crank and Flywheel Wet Vacuum Pumps

28″ Vacuum, with 30″ Barometer

Size of Pump			Diameter of Pump Openings					Displacement in Gallons per Minute, at 50 Revolutions per minute
Diam. Steam Cylinder	Diam. Pump Cylinder	Length of Stroke	Steam	Exhaust	Suction	Discharge	Charging Pipe (When Ordered)	
*8	10	12	1½	2	8	8	1	815
*8	12	12	1½	2	10	8	1¼	1175
*8	14	12	1½	2	10	8	1½	1600
8	10	8	2	2½	8	8	1	545
8	12	8	2	2½	8	8	1¼	780
10	12	10	2½	3½	8	8	1¼	980
10	14	10	2½	3½	10	10	1½	1330
10	14	12	2½	3½	10	10	1½	1600
10	15	12	2½	3½	10	10	1½	1835
10	16	12	2½	3½	12	12	2	2085
10	18	12	2½	3½	12	12	2	2645
12	16	12	3	4	12	12	2	2085
12	18	12	3	4	12	12	2	2645

* These sizes are side crank design.

Fig. 178



Duplex Crank and Flywheel Dry Vacuum Pumps

28½″-29″ Vacuum, 30″ Barometer

Size of Pump				Displacement Cubic Feet Free Air				Pipe Openings				
Diameter Steam Cylinder	Diameter Air Cylinder	Stroke	Revolutions Per Minute	Per Revolution	Per Minute	Required Steam Pressure	I. H. P. of Steam Cyl. at Peak Intake Atmospheric Disch.	Steam	Exhaust	Suction Each	Discharge Each	Cooling Water Inlet
8	18	8	250	4.70	1180	50	44	3	3½	5	5	¾
8	22	8	250	7.04	1748	90	62	3	3½	7	6	¾
10	18	8	250	4.70	1180	35	44	3½	4	5	5	¾
10	22	8	250	7.04	1748	50	62	3½	4	7	6	¾
10	22	10	235	8.80	2070	70	74	4	4½	7	6	1
10	26	10	235	12.30	2864	85	98	4	4½	8	7	1
12	22	10	235	8.80	2070	35	74	4½	5	7	6	1
12	26	10	235	12.30	2864	50	98	4½	5	8	7	1
12	28	12	220	17.12	3760	80	130	5	6	9	8	1
12	30	12	220	19.64	4320	100	150	5	6	10	8	1
14	28	12	220	17.12	3760	40	130	5	6	9	8	1
14	30	12	220	19.64	4320	45	150	5	6	10	8	1
14	32	15	210	27.96	5980	80	200	6	7	12	10	1
18	32	15	210	27.96	5980	35	200	7	8	12	10	1

Appendix

Creating a Vacuum in a Closed Tank

Quite often it is necessary to calculate the size of vacuum pump to exhaust a vessel of known capacity in a stated time to a certain degree of vacuum, and for this purpose the following table has been calculated. It gives the volume which must be exhausted from vessels in order to reduce the pressure from one atmosphere P to the lower pressure P_1. If the time is given in which a desired effect is to be produced, the size of pump can be readily calculated.

Table giving the number of cubic feet that must be exhausted from a closed vessel containing 100 to 4500 cubic feet in order to reduce the original internal pressure from (14.7 lbs.) to .9 - .01 atmospheres absolute or 3″ - 29 3-4″ vacuum.

The pressure in the vessel is to be reduced from atmosphere – P to – P_1		If the original pressure in a vessel is atmosphere absolute or P and it is to be reduced to P_1 the following volumes of air must be exhausted									
		Capacity of the vessel in cubic feet									
		100	500	1000	1500	2000	2500	3000	3500	4000	4500
Atmos. Abs.	Vac. Inch.	Cubic feet to be exhausted									
.9	3	10.5	53	105	158	210	263	315	368	424	473
.8	6	22.5	113	225	338	450	563	675	788	900	1013
.7	9	35	175	350	525	700	875	1050	1225	1400	1575
.6	12	51	255	510	765	1020	1275	1530	1785	2040	2295
.5	15	69	345	690	1035	1380	1725	2070	2415	2760	3106
.4	18	91.5	458	915	1374	1830	2290	2745	3203	3660	4118
.3	21	120	600	1200	1800	2400	3000	3600	4200	4800	5400
.25	22½	138	690	1380	2070	2760	3450	4140	4830	5520	6210
.2	24	161	805	1610	2415	3220	4025	4830	5635	6440	7245
.15	25½	190	950	1906	2850	3800	4750	5700	6650	7600	8550
.1	27	230	1150	2306	3450	4600	5750	6900	8050	9200	10350
.09	27¼	241	1205	2410	3615	4820	6025	7230	8435	9640	10845
.08	27½	252	1260	2520	3780	5040	6300	7560	8820	10080	11340
.07	27¾	266	1330	2660	3990	5320	6650	7980	9310	10640	11970
.06	28¼	281	1405	2810	4215	5620	7025	8430	9835	11240	12645
.05	28½	300	1500	3000	4500	6000	7500	9000	10500	12000	13500
.04	28¾	322	1610	3220	4830	6440	8050	9660	11270	12880	14490
.03	29	351	1755	3510	5265	7020	8775	10530	12285	14040	15795
.02	29½	391	1955	3910	5865	7820	9775	11730	13685	15640	17595
.01	29¾	460	2300	4600	6900	9200	11500	13800	16100	18400	20700

Example.

We have a closed tank of 500 cubic feet capacity at atmospheric pressure, and it is desired to exhaust it down to 21″ of vacuum in five minutes time. What capacity in cubic feet per minute must the air pump have?

Solution.

Referring to the table opposite 21″ of vacuum, it is seen for a vessel of 500 cubic feet capacity, 600 cubic feet must be exhausted. If this amount must be exhausted in five minutes time, the capacity of the air pump must be one-fifth of 600 or 120 cubic feet per minute.

Creating a Vacuum in a Closed Tank—Continued

If it is required to reduce the pressure in a vessel from P_2, which is lower than the atmosphere to the still lower pressure P_1, in order to calculate the volume of air to be exhausted, in this case, it is necessary to subtract the volume which must be exhausted in order to reduce the pressure from atmosphere to P_2, from that required to reduce the pressure from atmosphere to P_1.

EXAMPLE.

The vacuum in a closed tank of 2000 cubic feet capacity is 15″, and this is to be reduced to 27″ of vacuum. What volume must be exhausted?

SOLUTION.

From the table it is seen 4600 cubic feet must be exhausted to lower the pressure from atmosphere to 27″ of vacuum. Also it will be seen from the table 1380 cubic feet must be exhausted to lower the pressure from atmosphere to 15″ of vacuum. The difference between these two values equals 3220 cubic feet that must be exhausted to lower the pressure from 15″ of vacuum to 27″ of vacuum.

Having calculated the capacity required for the vacuum pump, the displacement of the vacuum pump must next be determined. This is calculated by assuming the volumetric efficiency of the vacuum pump as 60-75%. Then the displacement in cubic feet per minute equals the capacity in cubic feet per minute divided by the volumetric efficiency expressed as a decimal.

Properties of Saturated Steam

(Condensed from Marks and Davis's Steam Tables and Diagrams, 1909, by permission of the publishers, Longmans, Green & Co.)

Vacuum, Inches of Mercury.	Absolute Pressure, Lbs. per Sq. In.	Temperature, Fahrenheit.	Total Heat above 30° F.		Latent Heat, $L = H - h$ Heat-Units.	Volume, Cu. Ft. in 1 Lb. of Steam	Weight of 1 Cu. Ft. Steam, Lb.	Entropy of the Water.	Entropy of Evaporation.
			In the Water h Heat-Units.	In the Steam H Heat-Units.					
29.74	0.0886	32	0.00	1073.4	1073.4	3294	0.000304	0.0000	2.1832
29.67	0.1217	40	8.05	1076.9	1068.9	2438	0.000410	0.0162	2.1394
29.56	0.1780	50	18.08	1081.4	1063.3	1702	0.000587	0.0361	2.0865
29.40	0.2562	60	28.08	1085.9	1057.8	1208	0.000828	0.0555	2.0358
29.18	0.3626	70	38.06	1090.3	1052.3	871	0.001148	0.0745	1.9868
29.89	0.505	80	48.03	1094.8	1046.7	636.8	0.001570	0.0932	1.9398
28.50	0.696	90	58.00	1099.2	1041.2	469.3	0.002131	0.1114	1.8944
28.00	0.946	100	67.97	1103.6	1035.6	350.8	0.002851	0.1295	1.8505
27.88	1	101.83	69.8	1104.4	1034.6	333.0	0.00300	0.1327	1.8427
25.85	2	126.15	94.0	1115.0	1021.0	173.5	0.00576	0.1749	1.7431
23.81	3	141.52	109.4	1121.6	1012.3	118.5	0.00845	0.2008	1.6840
21.78	4	153.01	120.9	1126.5	1005.7	90.5	0.01107	0.2198	1.6416
19.74	5	162.28	130.1	1130.5	1000.3	73.33	0.01364	0.2348	1.6084
17.70	6	170.06	137.9	1133.7	995.8	61.89	0.01616	0.2471	1.5814
15.67	7	176.85	144.7	1136.5	991.8	53.56	0.01867	0.2579	1.5582
13.63	8	182.86	150.8	1139.0	988.2	47.27	0.02115	0.2673	1.5380
11.60	9	188.27	156.2	1141.1	985.0	42.36	0.02361	0.2756	1.5202
9.56	10	193.22	161.1	1143.1	982.0	38.38	0.02606	0.2832	1.5042
7.52	11	197.75	165.7	1144.9	979.2	35.10	0.02849	0.2902	1.4895
5.49	12	201.96	169.9	1146.5	976.6	32.36	0.03090	0.2967	1.4760
3.45	13	205.87	173.8	1148.0	974.2	30.03	0.03330	0.3025	1.4639
1.42	14	209.55	177.5	1149.4	971.9	28.02	0.03569	0.3081	1.4523
lbs. gage	14.70	212	180.0	1150.4	970.4	26.79	0.03732	0.3118	1.4447
0.3	15	213.0	181.0	1150.7	969.7	26.27	0.03806	0.3133	1.4416
1.3	16	216.3	184.4	1152.0	967.6	24.79	0.04042	0.3183	1.4311
2.3	17	219.4	187.5	1153.1	965.6	23.38	0.04277	0.3229	1.4215
3.3	18	222.4	190.5	1154.2	963.7	22.16	0.04512	0.3273	1.4127
4.3	19	225.2	193.4	1155.2	961.8	21.07	0.04746	0.3315	1.4045
5.3	20	228.0	196.1	1156.2	960.0	20.08	0.04980	0.3355	1.3965
6.3	21	230.6	198.8	1157.1	958.3	19.18	0.05213	0.3393	1.3887
7.3	22	233.1	201.3	1158.0	956.7	18.37	0.05445	0.3430	1.3811
8.3	23	235.5	203.8	1158.8	955.1	17.62	0.05676	0.3465	1.3739
9.3	24	237.8	206.1	1159.6	953.5	16.93	0.05907	0.3499	1.3670
10.3	25	240.1	208.4	1160.4	952.0	16.30	0.0614	0.3532	1.3604
11.3	26	242.2	210.6	1161.2	950.6	15.72	0.0636	0.3564	1.3542
12.3	27	244.4	212.7	1161.9	949.2	15.18	0.0659	0.3594	1.3483
13.3	28	246.4	214.8	1162.6	947.8	14.67	0.0682	0.3623	1.3425
14.3	29	248.4	216.8	1163.2	946.4	14.19	0.0705	0.3652	1.3367
15.3	30	250.3	218.8	1163.9	945.1	13.74	0.0728	0.3680	1.3311
16.3	31	252.2	220.7	1164.5	943.8	13.32	0.0751	0.3707	1.3257
17.3	32	254.1	222.6	1165.1	942.5	12.93	0.0773	0.3733	1.3205
18.3	33	255.8	224.4	1165.7	941.3	12.57	0.0795	0.3759	1.3155
19.3	34	257.6	226.2	1166.3	940.1	12.22	0.0818	0.3784	1.3107
20.3	35	259.3	227.9	1166.8	938.9	11.89	0.0841	0.3808	1.3060
21.3	36	261.0	229.6	1167.3	937.7	11.58	0.0863	0.3832	1.3014
22.3	37	262.6	231.3	1167.8	936.6	11.29	0.0886	0.3855	1.2969
23.3	38	264.2	232.9	1168.4	935.5	11.01	0.0908	0.3877	1.2925
24.3	39	265.8	234.5	1168.9	934.4	10.74	0.0931	0.3899	1.2882
25.3	40	267.3	236.1	1169.4	933.3	10.49	0.0953	0.3920	1.2841
26.3	41	268.7	237.6	1169.8	932.2	10.25	0.0976	0.3941	1.2800

Properties of Saturated Steam—Continued

Gauge Pressure, Lbs. per Sq. In.	Absolute Pressure, Lbs. per Sq. In.	Temperature, Fahrenheit.	Total Heat above 32° F.		Latent Heat, $L = H - h$ Heat-Units.	Volume, Cu. Ft. in 1 Lb. of Steam	Weight of 1 Cu. Ft. Steam, Lb.	Entropy of the Water.	Entropy of Evaporation.
			In the Water h Heat-Units.	In the Steam H Heat-Units.					
27.3	42	270.2	239.1	1170.3	931.2	10.02	0.0998	0.3962	1.2759
28.3	43	271.7	240.5	1170.7	930.2	9.80	0.1020	0.3982	1.2720
29.3	44	273.1	242.0	1171.2	929.2	9.59	0.1043	0.4002	1.2681
30.3	45	274.5	243.4	1171.6	928.2	9.39	0.1065	0.4021	1.2644
31 3	46	275.8	244.8	1172.0	927.2	9.20	0.1087	0.4040	1.2607
32.3	47	277.2	246.1	1172.4	926.3	9.02	0.1109	0.4059	1.2571
33.3	48	278.5	247.5	1172.8	925.3	8.84	0.1131	0.4077	1.2536
34.3	49	279.8	248.8	1173.2	924.4	8.67	0.1153	0.4095	1.2502
35.3	50	281.0	250.1	1173.6	923.5	8.51	0.1175	0.4113	1.2468
36.3	51	282.3	251.4	1174.0	922.6	8.35	0.1197	0.4130	1.2432
37.3	52	283.5	252.6	1174.3	921.7	8.20	0.1219	0.4147	1.2405
38.3	53	284.7	253.9	1174.7	920.8	8.05	0.1241	0.4164	1.2370
39.3	54	285.9	255.1	1175.0	919.9	7.91	0.1263	0.4180	1.2339
40.3	55	287.1	256.3	1175.4	919.0	7.78	0.1285	0.1496	1.2309
41.3	56	288.2	257.5	1175.7	918.2	7.65	0.1307	0.4212	1.2278
42.3	57	289.4	258.7	1176.0	917.4	7.52	0.1329	0.4227	1.2248
43.3	58	290.5	259.8	1176.4	916.5	7.40	0.1350	0.4242	1.2218
44.3	59	291.6	261.0	1176.7	915.7	7.28	0.1372	0.4257	1.2189
45.3	60	292.7	262.1	1177.0	914.9	7.17	0.1394	0.4272	1.2160
46.3	61	293.8	263.2	1177.3	914.1	7.06	0.1416	0.4287	1.2132
47.3	62	294.9	264.3	1177.6	913.3	6.95	0.1438	0.4302	1.2104
48.3	63	295.9	265.4	1177.9	912.5	6.85	0.1460	0.4316	1.2077
49.3	64	297.0	266.4	1178.2	911.8	6.75	0.1482	0.4330	1.2050
50.3	65	298.0	267.5	1178.5	911.0	6.65	0.1503	0.4344	1.2024
51.3	66	299.0	268.5	1178.8	910.2	6.56	0.1525	0.4358	1.1998
52.3	67	300.0	269.6	1179.0	909.5	6.47	0.1547	0.4371	1.1972
53.3	68	301.0	270.6	1179.3	908.7	6.38	0.1569	0.4385	1.1946
54.3	69	302.0	271.6	1179.6	908.0	6.29	0.1590	0.4398	1.1921
55.3	70	302.9	272.6	1179.8	907.2	6.20	0.1621	0.4411	1.1896
56.3	71	303.9	273.6	1180.1	906.5	6.12	0.1634	0.4424	1.1872
57.3	72	304.8	274.5	1180.4	905.8	6.04	0.1656	0.4437	1.1848
58.3	73	305.8	275.5	1180.6	905.1	5.96	0.1678	0.4449	1.1825
59.3	74	306.7	276.5	1180.9	904.4	5.89	0.1699	0.4462	1.1801
60.3	75	307.6	277.4	1181.1	903.7	5.81	0.1721	0.4474	1.1778
61.3	76	308.5	278.3	1181.4	903.0	5.74	0.1743	0.4487	1.1755
62.3	77	309.4	279.3	1181.6	902.3	5.67	0.1764	0.4499	1.1730
63.3	78	310.3	280.2	1181.8	901.7	5.60	0.1786	0.4511	1.1712
64.3	79	311.2	281.1	1182.1	901.0	5.54	0.1808	0.4523	1.1687
65.3	80	312.0	282.0	1182.3	900.3	5.47	0.1829	0.4535	1.1665
66.3	81	312.9	282.9	1182.5	899.7	5.41	0.1851	0.4546	1.1644
67.3	82	313.8	283.8	1182.8	899.0	5.34	0.1873	0.4557	1.1623
68.3	83	314.6	284.6	1183.0	898.4	5.28	0.1894	0.4568	1.1602
69.3	84	315.4	285.5	1183.2	897.7	5.22	0.1915	0.4579	1.1581
70.3	85	316.3	286.3	1183.4	897.1	5.16	0.1937	0.4590	1.1561
71.3	86	317.1	287.2	1183.6	896.4	5.10	0.1959	0.4601	1.1540
72.3	87	317.9	288.0	1183.8	895.8	5.05	0.1980	0.4612	1.1520
73.3	88	318.7	288.9	1184.0	895.2	5.00	0.2001	0.4623	1.1500
74.3	89	319.5	289.7	1184.2	894.6	4.94	0.2023	0.4633	1.1481
75.3	90	320.3	290.5	1184.4	893.9	4.89	0.2044	0.4644	1.1461
76.3	91	321.1	291.3	1184.6	893.3	4.84	0.2065	0.4654	1.1442
77.3	92	321.8	292.1	1184.8	892.7	4.79	0.2087	0.4664	1.1423
78.3	93	322.6	292.9	1185.0	892.1	4.74	0.2109	0.4674	1.1404
79.3	94	323.4	293.7	1185.2	891.5	4.69	0.2130	0.4684	1.1385
80.3	95	324.1	294.5	1185.4	890.9	4.65	0.2151	0.4694	1.1367

Properties of Saturated Steam—Continued

Gauge Pressure, Lbs. per Sq. In.	Absolute Pressure, Lbs. per Sq. In.	Temperature, Fahrenheit.	Total Heat above 32° F.		Latent Heat, $L = H - h$ Heat-Units.	Volume, Cu. Ft. in 1 Lb. of Steam	Weight of 1 Cu. Ft. Steam. Lb.	Entropy of the Water.	Entropy of Evaporation.
			In the Water h Heat-Units.	In the Steam H Heat Units.					
81.3	96	324.9	295.3	1185.6	890.3	4.60	0.2172	0.4704	1.1348
82.3	97	325.6	296.1	1185.8	889.7	4.56	0.2193	0.4714	1.1330
83.3	98	326.4	296.8	1186.0	889.2	4.51	0.2215	0.4724	1.1312
84.3	99	327.1	297.6	1186.2	888.6	4.47	0.2237	0.4733	1.1295
85.3	100	327.8	298.3	1186.3	888.0	4.429	0.2258	0.4743	1.1277
87.3	102	329.3	299.8	1186.7	886.9	4.347	0.2300	0.4762	1.1242
89.3	104	330.7	301.3	1187.0	885.8	4.268	0.2343	0.4780	1.1208
91.3	106	332.0	302.7	1187.4	884.7	4.192	0.2336	0.4798	1.1174
93.3	108	333.4	304.1	1187.7	883.6	4.118	0.2429	0.4816	1.1141
95.3	110	334.8	305.5	1188.0	882.5	4.047	0.2472	0.4834	1.1108
97.3	112	336.1	306.9	1188.4	881.4	3.978	0.2514	0.4852	1.1076
99.3	114	337.4	308.3	1188.7	880.4	3.912	0.2556	0.4869	1.1045
101.3	116	338.7	309.6	1189.0	879.3	3.848	0.2599	0.4886	1.1014
103.3	118	340.0	311.0	1189.3	878.3	3.786	0.2641	0.4903	1.0984
105.3	120	341.3	312.3	1189.6	877.2	3.726	0.2683	0.4919	1.0951
107.3	122	342.5	313.6	1189.8	876.2	3.668	0.2726	0.4935	1.0924
109.3	124	343.8	314.9	1190.1	875.2	3.611	0.2769	0.4951	1.0895
111.3	126	345.0	316.2	1190.4	874.2	3.556	0.2812	0.4967	1.0865
113.3	128	346.2	317.4	1190.7	873.3	3.504	0.2854	0.4982	1.0837
115.3	130	347.4	318.6	1191.0	872.3	3.452	0.2897	0.4998	1.0809
117.3	132	348.5	319.9	1191.2	871.3	3.402	0.2939	0.5013	1.0782
119.3	134	349.7	321.1	1191.5	870.4	3.354	0.2981	0.5028	1.0755
121.3	136	350.8	322.3	1191.7	869.4	3.308	0.3023	0.5043	1.0728
123.3	138	352.0	323.4	1192.0	868.5	3.263	0.3065	0.5057	1.0702
125.3	140	353.1	324.6	1192.2	867.6	3.219	0.3107	0.5072	1.0675
127.3	142	354.2	325.8	1192.5	866.7	3.175	0.3150	0.5086	1.0649
129.3	144	355.3	326.9	1192.7	865.8	3.133	0.3192	0.5100	1.0624
131.3	146	356.3	328.0	1192.9	864.9	3.092	0.3234	0.5114	1.0599
133.3	148	357.4	329.1	1193.2	864.0	3.052	0.3276	0.5128	1.0574
135.3	150	358.5	330.2	1193.4	863.2	3.012	0.3320	0.5142	1.0550
137.3	152	359.5	331.4	1193.6	862.3	2.974	0.3362	0.5155	1.0525
139.3	154	360.5	332.4	1193.8	861.4	2.938	0.3404	0.5169	1.0501
141.3	156	361.6	333.5	1194.1	860.6	2.902	0.3446	0.5182	1.0477
143.3	158	362.6	334.6	1194.3	859.7	2.868	0.3488	0.5195	1.0454
145.3	160	363.6	335.6	1194.5	858.8	2.834	0.3529	0.5208	1.0431
147.3	162	364.6	336.7	1194.7	858.0	2.801	0.3570	0.5220	1.0409
149.3	164	365.6	337.7	1194.9	857.2	2.769	0.3612	0.5233	1.0387
151.3	166	366.5	338.7	1195.1	856.4	2.737	0.3654	0.5245	1.0365
153.3	168	367.5	339.7	1195.3	855.5	2.706	0.3696	0.5257	1.0343
155.3	170	368.5	340.7	1195.4	854.7	2.675	0.3738	0.5269	1.0321
157.3	172	369.4	341.7	1195.6	853.9	2.645	0.3780	0.5281	1.0300
159.3	174	370.4	342.7	1195.8	853.1	2.616	0.3822	0.5293	1.0278
161.3	176	371.3	343.7	1196.0	852.3	2.588	0.3864	0.5305	1.0257
163.3	178	372.2	344.7	1196.2	851.5	2.560	0.3906	0.5317	1.0235
165.3	180	373.1	345.6	1196.4	850.8	2.533	0.3948	0.5328	1.0215
167.3	182	374.0	346.6	1196.6	850.0	2.507	0.3989	0.5339	1.0195
169.3	184	374.9	347.6	1196.8	849.2	2.481	0.4031	0.5351	1.0174
171.3	186	375.8	348.5	1196.9	848.4	2.455	0.4073	0.5362	1.0154
173.3	188	376.7	349.4	1197.1	847.7	2.430	0.4115	0.5373	1.0134
175.3	190	377.6	350.4	1197.3	846.9	2.406	0.4157	0.5384	1.0114
177.3	192	378.5	351.3	1197.4	846.1	2.381	0.4199	0.5395	1.0095
179.3	194	379.3	352.2	1197.6	845.4	2.358	0.4241	0.5405	1.0076
181.3	196	380.2	353.1	1197.8	844.7	2.335	0.4283	0.5416	1.0056
183.3	198	381.0	354.0	1197.9	843.9	2.312	0.4325	0.5426	1.0038

Properties of Saturated Steam—Continued

Gauge Pressure, Lbs. per Sq. In.	Absolute Pressure, Lbs. per Sq. In.	Temperature, Fahrenheit.	Total Heat above 32° F.		Latent Heat, L = H − h Heat-Units.	Volume, Cu. Ft. in 1 Lb. of Steam	Weight of 1 Cu. Ft. Steam, Lb.	Entropy of the Water.	Entropy of Evaporation.
			In the Water h Heat-Units.	In the Steam H Heat-Units.					
185.3	200	381.9	354.9	1198.1	843.2	2.290	0.437	0.5437	1.0019
190.3	205	384.0	357.1	1198.5	841.4	2.237	0.447	0.5463	0.9973
195.3	210	386.0	359.2	1198.8	839.6	2.187	0.457	0.5488	0.9928
200.3	215	388.0	361.4	1199.2	837.9	2.138	0.468	0.5513	0.9885
205.3	220	389.9	363.4	1199.6	836.2	2.091	0.478	0.5538	0.9841
210.3	225	391.9	365.5	1199.9	834.4	2.046	0.489	0.5562	0.9799
215.3	230	393.8	367.5	1200.2	832.8	2.004	0.499	0.5586	0.9758
220.3	235	395.6	369.4	1200.6	831.1	1.964	0.509	0.5610	0.9717
225.3	240	397.4	371.4	1200.9	829.5	1.924	0.520	0.5633	0.9676
230.3	245	399.3	373.3	1201.2	827.9	1.887	0.530	0.5655	0.9638
235.3	250	401.1	375.2	1201.5	826.3	1.850	0.541	0.5676	0.9600
245.3	260	404.5	378.9	1202.1	823.1	1.782	0.561	0.5719	0.9525
255.3	270	407.9	382.5	1202.6	820.1	1.718	0.582	0.5760	0.9454
265.3	280	411.2	385.0	1203.1	817.1	1.658	0.603	0.5800	0.9385
275.3	290	414.4	389.4	1203.6	814.2	1.602	0.624	0.5840	0.9316
285.3	300	417.5	392.7	1204.1	811.3	1.551	0.645	0.5878	0.9251
295.3	310	420.5	395.9	1204.5	808.5	1.502	0.666	0.5915	0.9187
305.3	320	423.4	399.1	1204.9	805.8	1.456	0.687	0.5951	0.9125
315.3	330	426.3	402.2	1205.3	803.1	1.413	0.708	0.5986	0.9065
325.3	340	429.1	405.3	1205.7	800.4	1.372	0.729	0.6020	0.9006
335.3	350	431.9	408.2	1206.1	797.8	1.334	0.750	0.6053	0.8949
345.3	360	434.6	411.2	1206.4	795.3	1.298	0.770	0.6085	0.8894
355.3	370	437.2	414.0	1206.8	792.8	1.264	0.791	0.6116	0.8840
365.3	380	439.8	416.8	1207.1	790.3	1.231	0.812	0.6147	0.8788
375.3	390	442.3	419.5	1207.4	787.9	1.200	0.833	0.6178	0.8737
385.3	400	444.8	422	1208	786	1.17	0.86	0.621	0.868
435.3	450	456.5	435	1209	774	1.04	0.96	0.635	0.844
485.3	500	467.3	448	1210	762	0.93	1.08	0.648	0.822
535.3	550	477.3	459	1210	751	0.83	1.20	0.659	0.801
585.3	600	486.6	469	1210	741	0.76	1.32	0.670	0.783

Properties of Superheated Steam

(Condensed from Marks and Davis's Steam Tables and Diagrams.)
v = specific volume in cu. ft. per lb., h = total heat, from water at 32° F. in B. T. U. per lb., n = entropy, from water at 32°.

Press. Abs. Lbs. per Sq. In.	Temp. Sat. Steam.		Degrees of Superheat.									
			0	20	50	100	150	200	250	300	400	500
20	228.0	v	20.08	20.73	21.69	23.25	24.80	26.33	27.85	29.37	32.39	35.40
		h	11562	1165.7	1179.9	1203.5	1227.1	1250.6	1274.1	1297.6	1344.8	1392.2
		n	1.7320	1.7456	1.7652	1.7961	1.8251	1.8524	1.8781	1.9026	1.9479	1.9893
40	267.3	v	10.49	10.83	11.33	12.13	12.93	13.70	14.48	15.25	16.78	18.30
		h	1169.4	1179.3	1194.0	1218.4	1242.4	1266.4	1290.3	1314.1	1361.6	1409.3
		n	1.6761	1.6895	1.7089	1.7392	1.7674	1.7940	1.8189	1.8427	1.8867	1.9271
60	292.7	v	7.17	7.40	7.75	8.30	8.84	9.36	9.89	10.41	11.43	12.45
		h	1177.0	1187.3	1202.6	1227.6	1252.1	1276.4	1300.4	1324.4	1372.2	1420.0
		n	1.6432	1.6568	1.6761	1.7062	1.7342	1.7603	1.7849	1.8081	1.8511	1.8908
80	312.0	v	5.47	5.65	5.92	6.34	6.75	7.17	7.56	7.95	8.72	9.49
		h	1182.3	1193.0	1208.8	1234.3	1259.0	1283.6	1307.8	1331.9	1379.8	1427.9
		n	1.6200	1.6338	1.6532	1.6833	1.7110	1.7368	1.7612	1.7840	1.8265	1.8658
100	327.8	v	4.43	4.58	4.79	5.14	5.47	5.80	6.12	6.44	7.07	7.69
		h	1186.3	1197.5	1213.8	1239.7	1264.7	1289.4	1313.6	1337.8	1385.9	1434.1
		n	1.6020	1.6160	1.6358	1.6658	1.6933	1.7188	1.7428	1.7656	1.8079	1.8468
120	341.3	v	3.73	3.85	4.04	4.33	4.62	4.89	5.17	5.44	5.96	6.48
		h	1189.6	1201.1	1217.9	1244.1	1269.3	1294.1	1318.4	1342.7	1391.0	1439.3
		n	1.5873	1.6016	1.6216	1.6517	1.6789	1.7041	1.7280	1.7505	1.7924	1.8311
140	353.1	v	3.22	3.32	3.49	3.75	4.00	4.24	4.48	4.71	5.16	5.61
		h	1192.2	1204.3	1221.4	1248.0	1273.3	1298.2	1322.6	1346.9	1395.4	1443.8
		n	1.5747	1.5894	1.6096	1.6395	1.6666	1.6916	1.7152	1.7376	1.7792	1.8177
160	363.6	v	2.83	2.93	3.07	3.30	3.53	3.74	3.95	4.15	4.56	4.95
		h	1194.5	1207.0	1224.5	1251.3	1276.8	1301.7	1326.2	1350.6	1399.3	1447.9
		n	1.5639	1.5789	1.5993	1.6292	1.6561	1.6810	1.7043	1.7266	1.7680	1.8063
180	373.1	v	2.53	2.62	2.75	2.96	3.16	3.35	3.54	3.72	4.09	4.44
		h	1196.4	1209.4	1227.2	1254.3	1279.9	1304.8	1329.5	1353.9	1402.7	1451.4
		n	1.5543	1.5697	1.5904	1.6201	1.6468	1.6716	1.6948	1.7169	1.7581	1.7962
200	381.9	v	2.29	2.37	2.49	2.68	2.86	3 04	3.21	3.38	3.71	4.03
		h	1198.1	1211.6	1229.8	1257.1	1282.6	1307.7	1332.4	1357.0	1405.9	1454.7
		n	1.5456	1.5614	1.5823	1.6120	1.6385	1.6632	1.6862	1.7082	1.7493	1.7872
220	389.9	v	2.09	2.16	2.28	2.45	2.62	2.78	2.94	3.10	3.40	3.69
		h	1199.6	1213.6	1232.2	1259.6	1285.2	1310.3	1335.1	1359.8	1408.8	1457.7
		n	1.5379	1.5541	1.5753	1.6049	1.6312	1.6558	1.6787	1.7005	1.7415	1.7792
240	397.4	v	1.92	1.99	2.09	2.26	2.42	2.57	2.71	2.85	3.13	3.40
		h	1200.9	1251.4	1234.3	1261.9	1287.6	1312.8	1337.6	1362.3	1411.5	1400.5
		n	1.5309	1.5476	1.5690	1.5985	1.6246	1.6492	1.6720	1.6937	1.7344	1.7721
260	404.5	v	1.78	1.84	1.94	2.10	2.24	2.39	2.52	2.65	2.91	3.16
		h	1202.1	1217.1	1236.4	1264.1	1289.9	1315.1	1340.0	1364.7	1414.0	1463.2
		n	1.5244	1.5416	1.5631	1.5926	1.6186	1.6430	1.6658	1.6874	1.7280	1.7655
280	411.2	v	1.66	1.72	1.81	1.95	2.09	2.22	2.35	2.48	2.72	2.95
		h	1203.1	1218.7	1238.4	1266.2	1291.9	1317.2	1342.2	1367.0	1416.4	1465.7
		n	1.5185	1.5362	1.5580	1.5873	1.6133	1.6375	1.6603	1.6818	1.7223	1.7597
300	417.5	v	1.55	1.60	1.69	1.83	1.96	2.09	2.21	2.33	2.55	2.77
		h	1204.1	1220.2	1240.3	1268.2	1294.0	1319.3	1344.3	1369.2	1418.6	1468.0
		n	1.5129	1.5310	1.5530	1.5824	1.6082	1.6323	1.6550	1.6765	1.7168	1.7541
350	431.9	v	1.33	1.38	1.46	1.58	1.70	1.81	1.92	2.02	2.22	2.41
		h	1206.1	1223.9	1244.6	1272.7	1298.7	1324.1	1349.3	1374.3	1424.0	1473.7
		n	1.5002	1.5199	1.5423	1.5715	1.5971	1.6210	1.6436	1.6650	1.7052	1.7422
400	444.8	v	1.17	1.21	1.28	1.40	1.50	1.60	1.70	1.79	1.97	2.14
		h	1207.7	1227.2	1248.6	1276.9	1303.0	1328.6	1353.9	1379.1	1429.0	1478.9
		n	1.4894	1.5107	1.5336	1.5625	1.5880	1.6117	1.6342	1.6554	1.6955	1.7323
450	456.5	v	1.04	1.08	1.14	1.25	1.35	1.44	1.53	1.61	1.77	1.93
		h	1209	1231	1252	1281	1307	1333	1358	1383	1434	1484
		n	1.479	1.502	1.526	1.554	1.580	1.603	1.626	1.647	1.687	1.723
500	467.3	v	0.93	0.97	1.03	1.13	1.22	1.31	1.39	1.47	1.62	1.76
		h	1210	1233	1256	1285	1311	1337	1362	1388	1438	1489
		n	1.470	1.496	1.519	1.548	1.573	1.597	1.619	1.640	1.679	1.715

Factors of Evaporation

(Hubbard's Steam Power Plants, Second Edition)

Temperature of Feed Water Degrees F	Steam Pressure by Guage																	
	50	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200	210	220
32	1.211	1.216	1.220	1.222	1.225	1.227	1.229	1.231	1.232	1.234	1.236	1.237	1.239	1.240	1.241	1.243	1.244	1.245
40	1.206	1.209	1.212	1.214	1.216	1.219	1.220	1.222	1.224	1.226	1.227	1.229	1.230	1.232	1.233	1.234	1.236	1.237
50	1.195	1.197	1.201	1.204	1.206	1.208	1.210	1.212	1.214	1.215	1.217	1.218	1.220	1.221	1.225	1.224	1.225	1.226
60	1.185	1.188	1.191	1.193	1.196	1.198	1.200	1.202	1.203	1.205	1.207	1.208	1.210	1.211	1.212	1.214	1.215	1.216
70	1.175	1.178	1.180	1.183	1.185	1.187	1.189	1.191	1.193	1.194	1.196	1.197	1.199	1.200	1.202	1.203	1.205	1.206
80	1.164	1.167	1.170	1.173	1.175	1.177	1.179	1.181	1.183	1.184	1.186	1.187	1.189	1.190	1.192	1.193	1.194	1.195
90	1.154	1.157	1.160	1.162	1.165	1.167	1.169	1.170	1.172	1.174	1.176	1.177	1.179	1.180	1.181	1.183	1.184	1.185
100	1.144	1.147	1.150	1.152	1.154	1.156	1.158	1.160	1.162	1.164	1.165	1.167	1.168	1.170	1.171	1.172	1.174	1.175
110	1.133	1.136	1.139	1.142	1.144	1 146	1.148	1.150	1.152	1.153	1.155	1.156	1.158	1.159	1.160	1.162	1.163	1.164
120	1.123	1.126	1.129	1.131	1.133	1.136	1.138	1.140	1.141	1.143	1.145	1.146	1.147	1.149	1.150	1.151	1.153	1.154
130	1.113	1.116	1.118	1.121	1.123	1.125	1.127	1.129	1.130	1.132	1.134	1.136	1.137	1.138	1.140	1.141	1.142	1.144
140	1.102	1.105	1.108	1.110	1.113	1.115	1.117	1.119	1.120	1.122	1.124	1.125	1.127	1.128	1.129	1.131	1.132	1.133
150	1.091	1.095	1.098	1.100	1.102	1.104	1.106	1.108	1.110	1.111	1.113	1.115	1.116	1.118	1.119	1.120	1.121	1.123
160	1.081	1.084	1.087	1.090	1.092	1.094	1.096	1.098	1.100	1.101	1.103	1.104	1.106	1.107	1.108	1.110	1.111	1.112
170	1.070	1.074	1.077	1.079	1.081	1.083	1.085	1.087	1.089	1.091	1.092	1.094	1.095	1.097	1.098	1.099	1.101	1.102
180	1.060	1.063	1.066	1.069	1.071	1.073	1.075	1.077	1.079	1.080	1.082	1.083	1.085	1.086	1.088	1.089	1.090	1.091
190	1.050	1.053	1.056	1.058	1.060	1.063	1.065	1.066	1.068	1.070	1.071	1.073	1.074	1.076	1.077	1.078	1.080	1.081
200	1.039	1.043	1.045	1.048	1.050	1.052	1.054	1.058	1.058	1.059	1.061	1.063	1.064	1.065	1.067	1.068	1.069	1.071
210	1.029	1.032	1.035	1.037	1.040	1.042	1.044	1.046	1.047	1.049	1.051	1.052	1.053	1.055	1.056	1.057	1.059	1.060

To find the equivalent evaporation from and at 212, multiply factor found in above table by actual water evaporated.

U. S. Standard

Schedule of Standard Flanges

1 inch to 40 inches, inclusive.
For Steam Pressures up to 125 lbs. per square inch.

Size of Pipe	Diameter of Flange	Thickness of Flange	Diameter of Bolt Circle	Number of Bolts	Size of Bolts	Diameter of Bolt Holes
1	4	7/16	3	4	7/16	9/16
1¼	4½	½	3⅜	4	7/16	9/16
1½	5	9/16	3⅞	4	½	⅝
2	6	⅝	4¾	4	⅝	¾
2½	7	11/16	5½	4	⅝	¾
3	7½	¾	6	4	⅝	¾
3½	8½	13/16	7	4	⅝	¾
4	9	15/16	7½	8	¾	¾
4½	9¼	15/16	7¾	8	⅝	⅞
5	10	15/16	8½	8	¾	⅞
6	11	1	9½	8	¾	⅞
7	12½	1 1/16	10¾	8	¾	⅞
8	13½	1⅛	11¾	8	¾	⅞
9	15	1⅛	13¼	12	¾	⅞
10	16	1 3/16	14¼	12	⅞	1
12	19	1¼	17	12	⅞	1
14	21	1⅜	18¾	12	1	1⅛
15	22¼	1⅜	20	16	1	1⅛
16	23½	1 7/16	21¼	16	1	1⅛
18	25	1 9/16	22¾	16	1⅛	1¼
20	27½	1 11/16	25	20	1⅛	1¼
22	29½	1 13/16	27¼	20	1¼	1⅜
24	32	1⅞	29½	20	1¼	1⅜
26	34¼	2	31¾	24	1¼	1⅜
28	36½	2 1/16	34	28	1¼	1⅜
30	38¾	2⅛	36	28	1⅜	1½
32	41¾	2¼	38½	28	1½	1⅝
34	43¾	2 5/16	40½	32	1½	1⅝
36	46	2⅜	42¾	32	1½	1⅝
38	48¾	2⅜	45¼	32	1⅝	1¾
40	50¾	2½	47¼	36	1⅝	1¾

U. S. Standard

Schedule of Extra Heavy Flanges

1 inch to 48 inches, inclusive
For Steam Pressures from 125 to 250 lbs. per square inch.

Size of Pipe	Diameter of Flange	Thickness of Flange	Diameter of Bolt Circle	Number of Bolts	Size of Bolts	Diameter of Bolt Holes
1	4 1/2	11/16	3 1/4	4	1/2	5/8
1 1/4	5	3/4	3 3/4	4	1/2	5/8
1 1/2	6	13/16	4 1/2	4	5/8	3/4
2	6 1/2	7/8	5	4	5/8	3/4
2 1/2	7 1/2	1	5 7/8	4	3/4	7/8
3	8 1/4	1 1/8	6 5/8	8	3/4	7/8
3 1/2	9	1 3/16	7 1/4	8	3/4	7/8
4	10	1 1/4	7 7/8	8	3/4	7/8
4 1/2	10 1/2	1 5/16	8 1/2	8	3/4	7/8
5	11	1 3/8	9 1/4	8	3/4	7/8
6	12 1/2	1 7/16	10 5/8	12	3/4	7/8
7	14	1 1/2	11 7/8	12	7/8	1
8	15	1 5/8	13	12	7/8	1
9	16 1/4	1 3/4	14	12	1	1 1/8
10	17 1/2	1 7/8	15 1/4	16	1	1 1/8
12	20 1/2	2	17 3/4	16	1 1/8	1 1/4
14	23	2 1/8	20 1/4	20	1 1/8	1 1/4
15	24 1/2	2 3/16	21 1/2	20	1 1/4	1 3/8
16	25 1/2	2 1/4	22 1/2	20	1 1/4	1 3/8
18	28	2 3/8	24 3/4	24	1 1/4	1 3/8
20	30 1/2	2 1/2	27	24	1 3/8	1 1/2
22	33	2 5/8	29 1/4	24	1 1/2	1 5/8
24	36	2 3/4	32	24	1 5/8	1 3/4
26	38 1/4	2 13/16	34 1/2	28	1 5/8	1 3/4
28	40 3/4	2 15/16	37	28	1 5/8	1 3/4
30	43	3	39 1/4	28	1 3/4	1 7/8
32	45 1/4	3 1/8	41 1/2	28	1 7/8	2
34	47 1/2	3 1/4	43 1/2	28	1 7/8	2
36	50	3 3/8	46	32	1 7/8	2
38	52 1/4	3 7/16	48	32	1 7/8	2
40	54 1/2	3 9/16	50 1/4	36	1 7/8	2
42	57	3 11/16	52 3/4	36	1 7/8	2
44	59 1/4	3 3/4	55	36	2	2 1/8
46	61 1/2	3 7/8	57 1/4	40	2	2 1/8
48	65	4	60 3/4	40	2	2 1/8

Table—Dimensions of Standard Wrought-Iron Pipe

(From Catalogue of Walworth Manufacturing Company)
FOR TEMPERATURES UP TO 450°
1¼ and smaller proved to 300 pounds per square inch by hydraulic pressure.
1½ and larger proved to 500 pounds per square inch by hydraulic pressure.

Nominal Diameter	Actual Outside Diameter	Actual Inside Diameter	Inside Circumference	Outside Circumference	Length of Pipe per Square Foot of inside Surface	Length of Pipe per Square Foot of outside Surface	Inside Area	Outside Area	Length of Pipe Containing one Cubic Foot	Weight per Foot
In.	In.	In.	In.	In.	Ft.	Ft.	In.	In.	Ft.	Lb.
⅛	0.405	0.270	0.848	1.272	14.15	9.44	0.0572	0.129	2500.	0.243
¼	0.54	0.364	1.144	1.696	10.50	7.075	0.1041	0.229	1385.	0.422
⅜	0.675	0.494	1.552	2.121	7.67	5.657	0.1916	0.358	751.5	0.561
½	0.84	0.623	1.957	2.652	6.13	4.502	0.3048	0.554	472.4	0.845
¾	1.05	0.824	2.589	3.299	4.635	3.637	0.5333	0.866	270.	1.126
1	1.315	1.048	3.292	4.134	3.679	2.903	0.8627	1.357	166.9	1.679
1¼	1.66	1.380	4.335	5.215	2.768	2.301	1.496	2.164	96.25	2.258
1½	1.90	1.611	5.061	5.969	2.371	2.01	2.038	2.835	70.65	2 694
2	2.375	2.067	6.494	7.461	1.848	1.611	3.355	4.430	42.36	3.600
2½	2.875	2.468	7.754	9.032	1.547	1.328	4.783	6.491	30.11	5.773
3	3.50	3.067	9.636	10.996	1.245	1.091	7.388	9.621	19.49	7.547
3½	4.00	3.548	11.146	12.566	1.077	0.955	9.887	12.566	14.56	9.055
4	4.50	4.026	12.648	14.137	0.949	0.849	12.730	15.904	11.31	10.66
4½	5.00	4.508	14.153	15.708	0.848	0.765	15.939	19.635	9.03	12.34
5	5.563	5.045	15.849	17.475	0.757	0.629	19.990	24.299	7.20	14.50
6	6.625	6.065	19.054	20.813	0.63	0.577	28.889	34.471	4.98	18.767
7	7.625	7.023	22.063	23.954	0.544	0.595	38 737	45.663	3.72	23.27
8	8.625	7.892	25.076	27.096	0.478	0.444	50.039	58.426	2.88	28.177
9	9.625	9.001	28.277	30 433	0.425	0.394	63.633	73.715	2.26	33.70
10	10.75	10.019	31.475	33.772	0.381	0.355	78.838	90.762	1.80	40.06
11	12 00	11.25	35.343	37.699	0.340	0.318	98.942	113.097	1.455	45.95
12	12.75	12.000	38.264	40.840	0.313	0.293	116.535	132.732	1.235	48.98
14	14.00	13.25	41.268	43.982	0.290	0.273	134.582	153.938	1.069	53.92
15	15.00	14.25	44.271	47.124	0 271	0.254	155.968	176.715	.923	57.89
16	16.00	15.25	47.274	50.265	0.254	0 238	177.867	201.062	.809	61.77
18	18.00	17.25	53.281	56.548	0.225	0.212	225.907	254.469	.638	69.66
20	20.00	19.25	59.288	62.832	0.202	0.191	279.720	314.160	.515	77.57

Extra Strong Pipe—Black and Galvanized

All Weights and Dimensions are Nominal

Size	Diameters		Thickness	Weight per foot Plain Ends
	External	Internal		
⅛	.405	.215	.095	.314
¼	.540	.302	.119	.535
⅜	.675	.423	.126	.738
½	.840	.546	.147	1.087
¾	1.050	.742	.154	1.473
1	1.315	.957	.179	2.171
1¼	1.660	1.278	.191	2.996
1½	1.900	1.500	.200	3.631
2	2.375	1.939	.218	5.022
2½	2.875	2.323	.276	7.661
3	3.500	2.900	.300	10.252
3½	4.000	3.364	.318	12.505
4	4.500	3.826	.337	14.983
4½	5.000	4.290	.355	17.611
5	5.563	4.813	.375	20.778
6	6.625	5.761	.432	28.573
7	7.625	6.625	.500	38.048
8	8.625	7.625	.500	43.388
9	9.625	8.625	.500	48.728
10	10.750	9.750	.500	54.735
11	11.750	10.750	.500	60.075
12	12.750	11.750	.500	65.415
13	14.000	13.000	.500	72.091
14	15.000	14.000	.500	77.431
15	16.000	15.000	.500	82.771

Double Extra Strong Pipe—Black and Galvanized

All Weights and Dimensions are Nominal

Size	Diameters		Thickness	Weight per foot Plain Ends
	External	Internal		
½	.840	.252	.294	1.714
¾	1.050	.434	.308	2.440
1	1.315	.599	.358	3.659
1¼	1.660	.896	.382	5.214
1½	1.900	1.100	.400	6.408
2	2.375	1.503	.436	9.029
2½	2.875	1.771	.552	13.695
3	3.500	2.300	.600	18.583
3½	4.000	2.728	.636	22.850
4	4.500	3.152	.674	27.541
4½	5.000	3.580	.710	32.530
5	5.563	4.063	.750	38.552
6	6.625	4.897	.864	53.160
7	7.625	5.875	.875	63.079
8	8.625	6.875	.875	72.424

Contents of Round Tanks in U. S. Gallons, for Each Foot in Depth

Diameter Ft.	In.	Gallons, 1 Foot in Depth	Diameter Ft.	In.	Gallons, 1 Foot in Depth	Diameter Ft.	In.	Gallons, 1 Foot in Depth
1	0	5.8735	11	0	710.6977	21	0	2590.2290
1	3	9.1766	11	3	743.3686	21	3	2652.2532
1	6	13.2150	11	6	776.7746	21	6	2715.0413
1	9	17.9870	11	9	810.9143	21	9	2778.5486
2	0	23.4940	12	0	848.1890	22	0	2842.7910
2	3	29.7340	12	3	881.3966	22	3	2907.7664
2	6	36.7092	12	6	917.7395	22	6	2973.4889
2	9	44.4179	12	9	954.8159	22	9	3039.9209
3	0	52.8618	13	0	992.6274	23	0	3107.1001
3	3	62.0386	13	3	1031.1719	23	3	3175.0122
3	6	73.1504	13	6	1070.4514	23	6	3243.6595
3	9	82.5959	13	9	1108.0645	23	9	3313.0403
4	0	93.9754	14	0	1151.2129	24	0	3383.1563
4	3	106.1290	14	3	1192.6940	24	3	3454.0051
4	6	118.9386	14	6	1234.9104	24	6	3525.5929
4	9	132.5209	14	9	1277.8615	24	9	3597.9068
5	0	146.8384	15	0	1321.5454	25	0	3670.9596
5	3	161.8886	15	3	1365.9634	25	3	3744.7452
5	6	177.6740	15	6	1407.5165	25	6	3819.2657
5	9	194.1913	15	9	1457.0032	25	9	3894.5203
6	0	211.4472	16	0	1503.6250	26	0	3970.5098
6	3	229.4342	16	3	1550.9797	26	3	4047.2322
6	6	248.1564	16	6	1599.0696	26	6	4124.6898
6	9	267.6122	16	9	1647.8930	26	9	4202.9610
7	0	287.8032	17	0	1697.4516	27	0	4281.8072
7	3	308.7270	17	3	1747.7431	27	3	4361.4664
7	6	330.3859	17	6	1798.7698	27	6	4441.8607
7	9	352.7665	17	9	1850.5301	27	9	4522.9886
8	0	375.9062	18	0	1903.0254	28	0	4604.8517
8	3	399.7666	18	3	1956.2537	28	3	4686.4876
8	6	424.3625	18	6	2010.2171	28	6	4770.7787
8	9	449.2118	18	9	2064.9140	28	9	4854.8434

Horse Power of an Engine

a = Area of the piston in square inches.
p = Mean effective pressure of the steam on the piston per square inch.
v = Velocity of piston per minute.

$$\text{Then H. P.} = \frac{a \times p \times v}{33{,}000}$$

The mean pressure in the cylinder when cutting off at

1/4	stroke	=	boiler	pressure	multiplied	by .597
1/3	"	=	"	"	"	" .670
3/8	"	=	"	"	"	" .743
1/2	"	=	"	"	"	" .847
5/8	"	=	"	"	"	" .919
2/3	"	=	"	"	"	" .937
3/4	"	=	"	"	"	" .966
7/8	"	=	"	"	"	" .992

To find the diameter of a cylinder of an engine of a required nominal horsepower:

$$\frac{5500}{v} \text{ multiplied by H. P.} = a.$$

Ranges in Steam Consumption by Prime Movers

Type Engine	Saturated Steam Lbs. per Hour	100° Super Lbs. per Hour	200° Super Lbs. per Hour
Simple—Non-condensing	29—45	20—38	18—35
Simple—Non-condensing automatic	26—40	18—34	16—30
Simple—Non-condensing Corliss	26—35	18—30	.....
Compound—Non-condensing	19—28	15—25	13—22
Compound—Condensing	12—22	10—20	9—17
Turbines Non-condensing (K. W. Hr.)	28—60	24—54	21—48
Turbines Condensing (K. W. Hr.)	12—42	10—38	9—34

Different Standards for Wire Gauge in Use in the United States

Dimensions of sizes in Decimal Parts of an Inch

Number of Wire Gauge	American or Brown & Sharpe	Birmingham or Stub's Wire	Washburn & Moen Mfg. Co. Worcester, Mass.	Imperial Wire Gauge	Stub's Steel Wire	U. S. Standard for Plate	Number of Wire Gauge
000000	.....	.....	.....	.464	.....	.46875	000000
00000	.....	.....	.....	.432	.....	.4375	00000
0000	.46	.454	.3938	.400	.....	.40625	0000
000	.40964	.425	.3625	.372	.....	.375	000
00	.3648	.38	.3310	.348	.....	.34375	00
0	.32486	.34	.3065	.324	.....	.3125	0
1	.2893	.3	.2830	.300	.227	.28125	1
2	.25763	.284	.2625	.276	.219	.265625	2
3	.22942	.259	.2437	.252	.212	.25	3
4	.20431	.238	.2253	.232	.207	.234375	4
5	.18194	.22	.2070	.212	.204	.21875	5
6	.16202	.203	.1920	.192	.201	.203125	6
7	.14428	.18	.1770	.176	.199	.1875	7
8	.12849	.165	.1620	.160	.197	.171875	8
9	.11443	.148	.1483	.144	.194	.15625	9
10	.10189	.134	.1350	.128	.191	.140625	10
11	.090742	.12	.1205	.116	.188	.125	11
12	.080808	.109	.1055	.104	.185	.109375	12
13	.071961	.095	.0915	.092	.182	.09375	13
14	.064084	.083	.0800	.080	.180	.078125	14
15	.057068	.072	.0720	.072	.178	.0708125	15
16	.05082	.065	.0625	.064	.175	.0625	16
17	.045257	.058	.0540	.056	.172	.05625	17
18	.040303	.049	.0475	.048	.168	.05	18
19	.03589	.042	.0410	.040	.164	.04375	19
20	.031961	.035	.0348	.036	.161	.0375	20
21	.028462	.032	.03175	.032	.157	.034375	21
22	.025347	.028	.0286	.028	.155	.03125	22
23	.022571	.025	.0258	.024	.153	.028125	23
24	.0201	.022	.0230	.022	.151	.025	24
25	.0179	.02	.0204	.020	.148	.021875	25
26	.01594	.018	.0181	.018	.146	.01875	26
27	.014195	.016	.0173	.0164	.143	.0171875	27
28	.012641	.014	.0162	.0149	.139	.015625	28
29	.011257	.013	.0150	.0136	.134	.0140625	29
30	.010025	.012	.0140	.0124	.127	.0125	30
31	.008928	.01	.0132	.0116	.120	.0109375	31
32	.00795	.009	.0128	.0108	.115	.01015625	32
33	.00708	.008	.0118	.0100	.112	.009375	33
34	.006304	.007	.0104	.0092	.110	.00859375	34
35	.005614	.005	.0095	.0084	.108	.0078125	35
36	.005	.004	.0090	.0076	.106	.00703125	36
37	.004453	.....	.....	.0068	.103	.006640625	37
38	.003965	.....	.....	.0060	.101	.00625	38
39	.003531	.....	.....	.0052	.099	.....	39
40	.003144	.....	.....	.0048	.097	.....	40

Logarithms of Numbers from 0 to 1000

No.	0	1	2	3	4	5	6	7	8	9
0	0	00000	30103	47712	60206	69897	77815	84510	90309	95424
10	00000	00432	00860	01284	01703	02119	02531	02938	03342	03743
11	04139	04532	04922	05308	05690	06070	06446	06819	07188	07555
12	07918	08279	08636	08991	09342	09691	10037	10380	10721	11059
13	11394	11727	12057	12385	12710	13033	13354	13672	13988	14301
14	14613	14922	15229	15534	15836	16137	16435	16732	17026	17319
15	17609	17898	18184	18469	18752	19033	19312	19590	19866	20140
16	20412	20683	20952	21219	21484	21748	22011	22272	22531	22789
17	23045	23300	23553	23805	24055	24304	24551	24797	25042	25285
18	25527	25768	26007	26245	26482	26717	26951	27184	27416	27646
19	27875	28103	28330	28556	28780	29003	29226	29447	29667	29885
20	30103	30320	30535	30750	30963	31175	31387	31597	31806	32015
21	32222	32428	32634	32838	33041	33244	33445	33646	33846	34044
22	34242	34439	34635	34830	35025	35218	35411	35603	35793	35984
23	36173	36361	36549	36736	36922	37107	37291	37475	37658	37840
24	38021	38202	38382	38561	38739	38917	39094	39270	39445	39620
25	39794	39967	40140	40312	40483	40654	40824	40993	41162	41330
26	41497	41664	41830	41996	42160	42325	42488	42651	42813	42975
27	43136	43297	43457	43616	43775	43933	44091	44248	44404	44560
28	44716	44871	45025	45179	45332	45484	45637	45788	45939	46090
29	46240	46389	46538	46687	46835	46982	47129	47276	47422	47567
30	47712	47857	48001	48144	48287	48430	48572	48714	48855	48996
31	49136	49276	49415	49554	49693	49831	49969	50106	50243	50379
32	50515	50651	50786	50920	51055	51188	51322	51455	51587	51720
33	51851	51983	52114	52244	52375	52504	52633	52763	52892	53020
34	53148	53275	53403	53529	53656	53782	53908	54033	54158	54283
35	54407	54531	54654	54777	54900	55023	55145	55267	55388	55509
36	55630	55751	55871	55991	56110	56229	56348	56467	56585	56703
37	56820	56937	57054	57171	57287	57403	57519	57634	57749	57864
38	57978	58093	58206	58320	58433	58546	58659	58771	58883	58995
39	59106	59218	59329	59439	59550	59660	59770	59879	59988	60097
40	60206	60314	60423	60531	60638	60746	60853	60959	61066	61172
41	61278	61384	61490	61595	61700	61805	61909	62014	62118	62221
42	62325	62428	62531	62634	62737	62839	62941	63043	63144	63246
43	63347	63448	63548	63649	63749	63849	63949	64048	64147	64246
44	64345	64444	64542	64640	64738	64836	64933	65031	65128	65225
45	65321	65418	65514	65610	65706	65801	65896	65992	66087	66181
46	66276	66370	66464	66558	66652	66745	66839	66932	67025	67117
47	67210	67302	67394	67486	67578	67669	67761	67852	67943	68034
48	68124	68215	68305	68395	68485	68574	68664	68753	68842	68931
49	69020	69108	69197	69285	69373	69461	69548	69636	69723	69810
50	69897	69984	70070	70157	70243	70329	70415	70501	70586	70672
51	70757	70842	70927	71012	71096	71181	71265	71349	71433	71517
52	71600	71684	71767	71850	71933	72016	72099	72181	72263	72346
53	72428	72509	72591	72673	72754	72835	72916	72997	73078	73159
54	73239	73320	73400	73480	73560	73640	73719	73799	73878	73957

Logarithms of Numbers, from 0 to 1000—Continued

No.	0	1	2	3	4	5	6	7	8	9
55	74036	74115	74194	74273	74351	74429	74507	74586	74663	74741
56	74819	74896	74974	75051	75128	75205	75282	75358	75435	75511
57	75587	75664	75740	75815	75891	75967	76042	76118	76193	76268
58	76343	76418	76492	76567	76641	76716	76790	76864	76938	77012
59	77085	77159	77232	77305	77379	77452	77525	77597	77670	77743
60	77815	77887	77960	78032	78104	78176	78247	78319	78390	78462
61	78533	78604	78675	78746	78817	78888	78958	79029	79099	79169
62	79239	79309	79379	79449	79518	79588	79657	79727	79796	79865
63	79934	80003	80072	80140	80209	80277	80346	80414	80482	80550
64	80618	80686	80754	80821	80889	80956	81023	81090	81158	81224
65	81291	81358	81425	81491	81558	81624	81690	81756	81823	81889
66	81954	82020	82086	82151	82217	82282	82347	82413	82478	82543
67	82607	82672	82737	82802	82866	82930	82995	83059	83123	83187
68	83251	83315	83378	83442	83506	83569	83632	83696	83759	83822
69	83885	83948	84011	84073	84136	84198	84261	84323	84386	84448
70	84510	84572	84634	84696	84757	84819	84880	84942	85003	85065
71	85126	85187	85248	85309	85370	85431	85491	85552	85612	85673
72	85733	85794	85854	85914	85974	86034	86094	86153	86213	86273
73	86332	86392	86451	86510	86570	86629	86688	86747	86806	86864
74	86923	86982	87040	87099	87157	87216	87274	87332	87390	87448
75	87506	87564	87622	87680	87737	87795	87852	87910	87967	88024
76	88081	88138	88196	88252	88309	88366	88423	88480	88536	88593
77	88649	88705	88762	88818	88874	88930	88986	89042	89098	89154
78	89209	89265	89321	89376	89432	89487	89542	89597	89653	89708
79	89763	89818	89873	89927	89982	90037	90091	90146	90200	90255
80	90309	90363	90417	90472	90526	90580	90634	90687	90741	90795
81	90849	90902	90956	91009	91062	91116	91169	91222	91275	91328
82	91381	91434	91487	91540	91593	91645	91698	91751	91803	91855
83	91908	91960	92012	92065	92117	92169	92221	92273	92324	92376
84	92428	92480	92531	92583	92634	92686	92737	92788	92840	92891
85	92942	92993	93044	93095	93146	93197	93247	93298	93349	93399
86	93450	93500	93551	93601	93651	93702	93752	93802	93852	93902
87	93952	94002	94052	94101	94151	94201	94250	94300	94349	94399
88	94448	94498	94547	94596	94645	94694	94743	94792	94841	94890
89	94939	94988	95036	95085	95134	95182	95231	95279	95328	95376
90	95424	95472	95521	95569	95617	95665	95713	95761	95809	95856
91	95904	95952	95999	96047	96095	96142	96190	96237	96284	96332
92	96379	96426	96473	96520	96567	96614	96661	96708	96755	96802
93	96848	96895	96942	96988	97035	97081	97128	97174	97220	97267
94	97313	97359	97405	97451	97497	97543	97589	97635	97681	97727
95	97772	97818	97864	97909	97955	98000	98046	98091	98137	98182
96	98227	98272	98318	98363	98408	98453	98498	98543	98588	98632
97	98677	98722	98767	98811	98856	98900	98945	98989	99034	99078
98	99123	99167	99211	99255	99300	99344	99388	99432	99476	99520
99	99564	99607	99651	99695	99739	99782	99826	99870	99913	99957

Natural Trigonometrical Functions

°	M.	Sine.	Co-vers.	Co-sec.	Tang.	Co-tan.	Secant.	Ver.Sin.	Co-sine		
0	0	.00000	1.00000	Infinite	.00000	Infinite	1.0000	.00000	1.00000	90	0
	15	.00436	.99564	229.18	.00436	229.18	1.0000	.00001	.99999		45
	30	.00873	.99127	114.59	.00873	114.59	1.0000	.00004	.99996		30
	45	.01309	.98691	76.397	.01309	76.390	1.0001	.00009	.99991		15
1	0	.01745	.98255	57.299	.01745	57.290	1.0001	.00015	.99985	89	0
	15	.02181	.97819	45.840	.02182	45.829	1.0002	.00024	.99976		45
	30	.02618	.97382	38.202	.02618	38.188	1.0003	.00034	.99966		30
	45	.03054	.96946	32.746	.03055	32.730	1.0005	.00047	.99953		15
2	0	.03490	.96510	28.654	.03492	28.636	1.0006	.00061	.99939	88	0
	15	.03926	.96074	25.471	.03929	25.452	1.0008	.00077	.99923		45
	30	.04362	.95638	22.926	.04366	22.904	1.0009	.00095	.99905		30
	45	.04798	.95202	20.843	.04803	20.819	1.0011	.00115	.99885		15
3	0	.05234	.94766	19.107	.05241	19.081	1.0014	.00137	.99863	87	0
	15	.05669	.94331	17.639	.05678	17.611	1.0016	.00161	.99839		45
	30	.06105	.93895	16.380	.06116	16.350	1.0019	.00187	.99813		30
	45	.06540	.93460	15.290	.06554	15.527	1.0021	.00214	.99786		15
4	0	.06976	.93024	13.336	.06993	14.301	1.0024	.00244	.99756	86	0
	15	.07411	.92589	13.494	.07431	13.457	1.0028	.00275	.99725		45
	30	.07846	.92154	12.745	.07870	12.706	1.0031	.00308	.99692		30
	45	.08281	.91719	12.076	.08309	12.035	1.0034	.00343	.99656		15
5	0	.08716	.91284	11.474	.08749	11.430	1.0038	.00381	.99619	85	0
	15	.09150	.90850	10.929	.09189	10.883	1.0042	.00420	.99580		45
	30	.09585	.90415	10.433	.09629	10.385	1.0046	.00460	.99540		30
	45	.10019	.89981	9.9812	.10069	9.9310	1.0051	.00503	.99497		15
6	0	.10453	.89547	9.5668	.10510	9.5144	1.0055	.00548	.99452	84	0
	15	.10887	.89113	9.1855	.10952	9.1309	1.0060	.00594	.99406		45
	30	.11320	.88680	8.8337	.11393	8.7769	1.0065	.00643	.99357		30
	45	.11754	.88246	8.5079	.11836	8.4490	1.0070	.00693	.99307		15
7	0	.12187	.87813	8.2055	.12278	8.1443	1.0075	.00745	.99255	83	0
	15	.12620	.87380	7.9240	.12722	7.8606	1.0081	.00800	.99200		45
	30	.13053	.86947	7.6613	.13165	7.5958	1.0086	.00856	.99144		30
	45	.13485	.86515	7.4156	.13609	7.3479	1.0092	.00913	.99086		15
8	0	.13917	.86083	7.1853	.14054	7.1154	1.0098	.00973	.99027	82	0
	15	.14349	.85651	6.9690	.14199	6.8969	1.0105	.01035	.98965		45
	30	.14781	.85219	6.7655	.14945	6.6912	1.0111	.01098	.98902		30
	45	.15212	.84788	6.5736	.15391	6.4971	1.0118	.01164	.98836		15
9	0	.15643	.84357	6.3924	.15838	6.3138	1.0125	.01231	.98769	81	0
	15	.16074	.83926	6.2211	.16286	6.1402	1.0132	.01300	.98700		45
	30	.16505	.83495	6.0589	.16734	5.9758	1.0139	.01371	.98629		30
	45	.16935	.83065	5.9049	.17183	5.8197	1.0147	.01444	.98556		15
10	0	.17365	.82635	5.7588	.17633	5.6713	1.0154	.01519	.98481	80	0
	15	.17794	.82206	5.6198	.18083	5.5301	1.0162	.01596	.98404		45
	30	.18224	.81776	5.4874	.18534	5.3955	1.0170	.01675	.98325		30
	45	.18652	.81348	5.3612	.18986	5.2672	1.0179	.01755	.98245		15
11	0	.19081	.80919	5.2408	.19438	5.1446	1.0187	.01837	.98163	79	0
	15	.19509	.80491	5.1258	.19891	5.0273	1.0196	.01921	.98079		45
	30	.19937	.80063	5.0158	.20345	4.9152	1.0205	.02008	.97992		30
	45	.20364	.79636	4.9106	.20800	4.8077	1.0214	.02095	.97905		15
12	0	.20791	.79209	4.8097	.21256	4.7046	1.0223	.02185	.97815	78	0
	15	.21218	.78782	4.7130	.21712	4.6057	1.0233	.02277	.97723		45
	30	.21644	.78356	4.6202	.22169	4.5107	1.0243	.02370	.97630		30
	45	.22070	.77930	4.5311	.22628	4.4194	1.0253	.02466	.97534		15
13	0	.22495	.77505	4.4454	.23087	4.3315	1.0263	.02563	.97437	77	0
	15	.22920	.77080	4.3630	.23547	4.2468	1.0273	.02662	.97338		45
	30	.23345	.76655	4.2837	.24008	4.1653	1.0284	.02763	.97237		30
	45	.23769	.76231	4.2072	.24470	4.0867	1.0295	.02866	.97134		15
14	0	.24192	.75808	4.1336	.24933	4.0108	1.0306	.02970	.97030	76	0
	15	.24615	.75385	4.0625	.25397	3.9375	1.0317	.03077	.96923		45
	30	.25038	.74962	3.9939	.25862	3.8667	1.0329	.03185	.96815		30
	45	.25460	.74540	3.9277	.26328	3.7983	1.0341	.03295	.96705		15
15	0	.25882	.74118	3.8637	.26795	3.7320	1.0353	.03407	.96593	75	0
		Co-sine.	Ver.Sin.	Secant.	Co-tan.	Tang.	Co-sec.	Co-vers.	Sine.	°	M

From 75° to 90° read from bottom of table upwards.

Natural Trigonometrical Functions—Continued

°	M.	Sine.	Co-vers.	Co-sec.	Tang.	Co-tan.	Secant.	Ver.Sin.	Co-sine		
15	0	.25882	.74118	3.8637	.26795	3.7320	1.0353	.03407	.96593	75	0
	15	.26303	.73697	3.8018	.27263	3.6680	1.0365	.03521	.96479		45
	30	.26724	.73276	3.7420	.27732	3.6059	1.0377	.03637	.96363		30
	45	.27144	.72856	3.6840	.28203	3.5457	1.0390	.03754	.96246		15
16	0	.27564	.72436	3.6280	.28674	3.4874	1.0403	.03874	.96126	74	0
	15	.27983	.72017	3.5736	.29147	3.4308	1.0416	.03995	.96005		45
	30	.28402	.71598	3.5209	.29621	3.3759	1.0429	.04118	.95882		30
	45	.28820	.71180	3.4699	.30096	3.3226	1.0443	.04243	.95757		15
17	0	.29237	.70763	3.4203	.30573	3.2709	1.0457	.04370	.95630	73	0
	15	.29654	.70346	3.3722	.31051	3.2205	1.0471	.04498	.95502		45
	30	.30070	.69929	3.3255	.31530	3.1716	1.0485	.04628	.95372		30
	45	.30486	.69514	3.2801	.32010	3.1240	1.0500	.04760	.95240		15
18	0	.30902	.69908	3.2361	.32492	3.0777	1.0515	.04894	.95106	72	0
	15	.31316	.68684	3.1932	.32975	3.0326	1.0530	.05030	.94970		45
	30	.31730	.68270	3.1515	.33459	2.9887	1.0545	.05168	.94832		30
	45	.32144	.67856	3.1110	.33945	2.9459	1.0560	.05307	.94693		15
19	0	.32557	.67443	3.0715	.34433	2.9042	1.0576	.05448	.94552	71	0
	15	.32969	.67031	3.0331	.34921	2.8636	1.0592	.05591	.94409		45
	30	.33381	.66619	2.9957	.35412	2.8239	1.0608	.05736	.94264		30
	45	.33792	.66208	2.9593	.35904	2.7852	1.0625	.05882	.94118		15
20	0	.34202	.65798	2.9238	.36397	2.7475	1.0642	.06031	.93969	70	0
	15	.34612	.65388	2.8892	.36892	2.7106	1.0659	.06181	.93819		45
	30	.35021	.64979	2.8554	.37388	2.6746	1.0676	.06333	.93667		30
	45	.35429	.64571	2.8225	.37887	2.6395	1.0694	.06486	.93514		15
21	0	.35837	.64163	2.7904	.38386	2.6051	1.0711	.06642	.93358	69	0
	15	.36244	.63756	2.7591	.38888	2.5715	1.0729	.06799	.93201		45
	30	.36650	.63350	2.7285	.39391	2.5386	1.0748	.06958	.93042		30
	45	.37056	.62944	2.6986	.39896	2.5065	1.0766	.07119	.92881		15
22	0	.37461	.62539	2.6695	.40403	2.4751	1.0785	.07282	.92718	68	0
	15	.37865	.62135	2.6410	.40911	2.4443	1.0804	.07446	.92554		45
	30	.38268	.61732	2.6131	.41421	2.4142	1.0824	.07612	.92388		30
	45	.38679	.61329	2.5859	.41933	2.3847	1.0844	.07780	.92220		15
23	0	.39073	.60927	2.5593	.42447	2.3559	1.0864	.07950	.92050	67	0
	15	.39474	.60526	2.5333	.42963	2.3276	1.0884	.08121	.91879		45
	30	.39875	.60125	2.5078	.43481	2.2998	1.0904	.08294	.91706		30
	45	.40275	.59725	2.4829	.44001	2.2727	1.0925	.08469	.91531		15
24	0	.40674	.59326	2.4586	.44523	2.2460	1.0946	.08645	.91355	66	0
	15	.41072	.58928	2.4348	.45047	2.2199	1.0968	.08824	.91176		45
	30	.41469	.58531	2.4114	.45573	2.1943	1.0989	.09004	.90996		30
	45	.41866	.58134	2.3886	.46101	2.1692	1.1011	.09186	.90814		15
25	0	.42262	.57738	2.3662	.46631	2.1445	1.1034	.09369	.90631	65	0
	15	.42657	.57343	2.3443	.47163	2.1203	1.1056	.09554	.90446		45
	30	.43051	.56949	2.3228	.47697	2.0965	1.1079	.09741	.90259		30
	45	.43445	.56555	2.3018	.48234	2.0732	1.1102	.09930	.90070		15
26	0	.43837	.56163	2.2812	.48773	2.0503	1.1126	.10121	.89879	64	0
	15	.44229	.55771	2.2610	.49314	2.0278	1.1150	.10313	.89687		45
	30	.44620	.55380	2.2412	.49858	2.0057	1.1174	.10507	.89493		30
	45	.45010	.54990	2.2217	.50404	1.9840	1.1198	.10702	.89298		15
27	0	.45399	.54601	2.2027	.50952	1.9626	1.1223	.10899	.89101	63	0
	15	.45787	.54213	2.1840	.51503	1.9416	1.1248	.11098	.88902		45
	30	.46175	.53825	2.1657	.52057	1.9210	1.1274	.11299	.88701		30
	45	.46561	.53439	2.1477	.52612	1.9007	1.1300	.11501	.88499		15
28	0	.46947	.53053	2.1300	.53171	1.8807	1.1326	.11705	.88295	62	0
	15	.47332	.52668	2.1127	.53732	1.8611	1.1352	.11911	.88089		45
	30	.47716	.52284	2.0957	.54295	1.8418	1.1379	.12118	.87882		30
	45	.48099	.51901	2.0790	.54862	1.8228	1.1406	.12327	.87673		15
29	0	.48481	.51519	2.0627	.55431	1.8040	1.1433	.12538	.87462	61	0
	15	.48862	.51138	2.0466	.56003	1.7856	1.1461	.12750	.87250		45
	30	.49242	.50758	2.0308	.56577	1.7675	1.1490	.12964	.87036		30
	45	.49622	.50378	2.0152	.57155	1.7496	1.1518	.13180	.86820		15
30	0	.50000	.50000	2.0000	.57735	1.7320	1.1547	.13397	.86603	60	0
		Co-sine.	Ver.Sin.	Secant.	Co-tan.	Tang.	Co-sec.	Co-vers.	Sine.	°	M

From 60° to 75° read from bottom of table upwards.

Natural Trigonometrical Functions—Continued

°	M.	Sine.	Co-vers.	Co-sec.	Tang.	Co-tan.	Secant.	Ver.Sin.	Co-sine		
30	0	.50000	.50000	2.0000	.57735	1.7320	1.1547	.13397	.86603	60	0
	15	.50377	.49623	1.9850	.58318	1.7147	1.1576	.13616	.86384		45
	30	.50754	.49246	1.9703	.58904	1.6977	1.1606	.13837	.86163		30
	45	.51129	.48871	1.9558	.59494	1.6808	1.1636	.14059	.85941		15
31	0	.51504	.48496	1.9416	.60086	1.6643	1.1666	.14283	.85717	59	0
	15	.51877	.48123	1.9276	.60681	1.6479	1.1697	.14509	.85491		45
	30	.52250	.47750	1.9139	.61280	1.6319	1.1728	.14736	.85264		30
	45	.52621	.47379	1.9004	.61882	1.6160	1.1760	.14965	.85035		15
32	0	.52992	.47008	1.8871	.62487	1.6003	1.1792	.15195	.84805	58	0
	15	.53361	.46639	1.8740	.63095	1.5849	1.1824	.15427	.84573		45
	30	.53730	.46270	1.8612	.63707	1.5697	1.1857	.15661	.84339		30
	45	.54097	.45903	1.8485	.64322	1.5547	1.1890	.15896	.84104		15
33	0	.54464	.45536	1.8361	.64941	1.5399	1.1924	.16733	.83867	57	0
	15	.54829	.45171	1.8238	.65563	1.5253	1.1958	.16371	.83629		45
	30	.55194	.44806	1.8118	.66188	1.5108	1.1992	.16611	.83389		30
	45	.55557	.44443	1.7999	.66818	1.4966	1.2027	.16853	.83147		15
34	0	.55919	.44081	1.7883	.67451	1.4826	1.2062	.17096	.82904	56	0
	15	.56280	.43720	1.7768	.68807	1.4687	1.2098	.17341	.82659		45
	30	.56641	.43359	1.7655	.63728	1.4550	1.2134	.17587	.82413		30
	45	.57000	.43000	1.7544	.69372	1.4415	1.2171	.17835	.82165		15
35	0	.57358	.42642	1.7434	.70021	1.4281	1.2208	.18085	.81915	55	0
	15	.57715	.42285	1.7327	.70673	1.4150	1.2245	.18336	.81664		45
	30	.58070	.41930	1.7220	.71329	1.4019	1.2283	.18588	.81412		30
	45	.58425	.41575	1.7116	.71990	1.3891	1.2322	.18843	.81157		15
36	0	.58779	.41221	1.7013	.72654	1.3764	1.2361	.19098	.80902	54	0
	15	.59131	.40869	1.6912	.73323	1.3638	1.2400	.19356	.80644		45
	30	.59482	.40518	1.6812	.73996	1.3514	1.2440	.19614	.80386		30
	45	.59832	.40168	1.6713	.74673	1.3392	1.2480	.19875	.80125		15
37	0	.60181	.39819	1.6616	.75355	1.3270	1.2521	.20136	.79864	53	0
	15	.60529	.39471	1.6521	.76042	1.3151	1.2563	.20400	.79600		45
	30	.60876	.39124	1.6427	.76733	1.3032	1.2605	.20665	.79335		30
	45	.61222	.38778	1.6334	.77428	1.2915	1.2647	.20931	.79069		15
38	0	.61566	.38434	1.6243	.78129	1.2799	1.2690	.21199	.78801	52	0
	15	.61909	.38091	1.6153	.78834	1.2685	1.2734	.21468	.78532		45
	30	.62251	.37749	1.6064	.79543	1.2572	1.2778	.21739	.78261		30
	45	.62592	.37408	1.5976	.80258	1.2460	1.2822	.22012	.77988		15
39	0	.62932	.37068	1.5890	.80978	1.2349	1.2868	.22285	.77715	51	0
	15	.63271	.36729	1.5805	.81703	1.2239	1.2913	.22561	.77439		45
	30	.63608	.36392	1.5721	.82434	1.2131	1.2960	.22838	.77162		30
	45	.63944	.36056	1.5639	.83169	1.2024	1.3007	.23116	.76884		15
40	0	.64279	.35721	1.5557	.83910	1.1918	1.3054	.23396	.76604	50	0
	15	.64612	.35388	1.5477	.84656	1.1812	1.3102	.23677	.76323		45
	30	.64945	.35055	1.5398	.85408	1.1708	1.3151	.23959	.76041		30
	45	.65276	.34724	1.5320	.86165	1.1606	1.3200	.24244	.75756		15
41	0	.65606	.34394	1.5242	.86929	1.1504	1.3250	.24529	.75471	49	0
	15	.65935	.34065	1.5166	.87698	1.1403	1.3301	.24816	.75184		45
	30	.66262	.33738	1.5092	.88472	1.1303	1.3352	.25104	.74896		30
	45	.66588	.33412	1.5018	.89253	1.1204	1.3404	.25394	.74606		15
42	0	.66913	.33087	1.4945	.90040	1.1106	1.3456	.25686	.74314	48	0
	15	.67237	.32763	1.4873	.90834	1.1009	1.3509	.25978	.74022		45
	30	.67559	.32441	1.4802	.91633	1.0913	1.3563	.26272	.73728		30
	45	.67880	.32120	1.4732	.92439	1.0818	1.3618	.26568	.73432		15
43	0	.68200	.31800	1.4663	.93251	1.0724	1.3673	.26865	.73135	47	0
	15	.68518	.31482	1.4595	.94071	1.0630	1.3729	.27163	.72837		45
	30	.68835	.31165	1.4527	.94896	1.0538	1.3786	.27463	.72537		30
	45	.69151	.30849	1.4461	.95729	1.0446	1.3843	.27764	.72236		15
44	0	.69466	.30534	1.4396	.96569	1.0355	1.3902	.28066	.71934	46	0
	15	.69779	.30221	1.4331	.97416	1.0265	1.3961	.28370	.71630		45
	30	.70091	.29909	1.4267	.98270	1.0176	1.4020	.28675	.71325		30
	45	.70401	.29599	1.4204	.99131	1.0088	1.4081	.28981	.71019		15
45	0	.70711	.29289	1.4142	1.0000	1.0000	1.4142	.29289	.70711	45	0
		Co-sine.	Ver. Sin.	Secant.	Co-tan.	Tang.	Co-sec.	Co-vers.	Sine.	°	M.

From 40° to 60° read from bottom of table upwards:

Useful Information—Comparison of Thermometers

Cent.	Reau.	Fahr.	Cent.	Reau.	Fahr.	Cent.	Reau.	Fahr.
—40	—32.0	—40.0	21	16.8	69.8	62	49.6	143.6
—38	—30.4	—36.4	22	17.6	71.6	63	50.4	145.4
—36	—28.8	—32.8	23	18.4	73.4	64	51.2	147.2
—34	—27.2	—29.2	24	19.2	75.2	65	52.0	149.0
—32	—25.6	—25.6	25	20.0	77.0	66	52.8	150.8
—30	—24.0	—22.0	26	20.8	78.8	67	53.6	152.6
—28	—22.4	—18.4	27	21.6	80.6	68	54.4	154.4
—26	—20.8	—14.8	28	22.4	82.4	69	55.2	156.2
—24	—19.2	—11.2	29	23.2	84.2	70	56.0	158.0
—22	—17.6	— 7.6	30	24.0	86.0	71	56.8	159.8
—20	—16.0	— 4.0	31	24.8	87.8	72	57.6	161.6
—18	—14.4	— 0.4	32	25.6	89.6	73	58.4	163.4
—16	—12.8	+ 3.2	33	26.4	91.4	74	59.2	165.2
—14	—11.2	6.8	34	27.2	93.2	75	60.0	167.0
—12	— 9.6	10.4	35	28.0	95.0	76	60.8	168.8
—10	— 8.0	14.0	36	28.8	96.8	77	61.6	170.6
— 8	— 6.4	17.6	37	29.6	98.6	78	62.4	172.4
— 6	— 4.8	21.2	38	30.4	100.4	79	63.2	174.2
— 4	— 3.2	24.8	39	31.2	102.2	80	64.0	176.0
— 2	— 1.6	28.4	40	32.0	104.0	81	64.8	177.8
0	0.0	32.0	41	32.8	105.8	82	65.6	179.6
+ 1	+ 0.8	33.8	42	33.6	107.6	83	66.4	181.4
2	1.6	35.6	43	34.4	109.4	84	67.2	183.2
3	2.4	37.4	44	35.2	111.2	85	68.0	185.0
4	3.2	39.2	45	36.0	113.0	86	68.8	186.8
5	4.0	41.0	46	36.8	114.8	87	69.6	188.6
6	4.8	42.8	47	37.6	116.6	88	70.4	190.4
7	5.6	44.6	48	38.4	118.4	89	71.2	192.2
8	6.4	46.4	49	39.2	120.2	90	72.0	194.0
9	7.2	48.2	50	40.0	122.0	91	72.8	195.8
10	8.0	50.0	51	40.8	123.8	92	73.6	197.6
11	8.8	51.8	52	41.6	125.6	93	74.4	199.4
12	9.6	53.6	53	42.4	127.4	94	75.2	201.2
13	10.4	55.5	54	43.2	129.2	95	76.0	203.0
14	11.2	57.2	55	44.0	131.0	96	76.8	204.8
15	12.0	59.0	56	44.8	132.8	97	77.6	206.6
16	12.8	60.8	57	45.6	134.6	98	78.4	208.4
17	13.6	62.6	58	46.4	136.4	99	79.2	210.2
18	14.4	64.4	59	47.2	138.2	100	80.0	212.0
19	15.2	66.2	60	48.0	140.0			
20	16.0	68.0	61	48.8	141.8			

Freezing point on Fahrenheit scale is +32 degrees: boiling point, 212 degrees.

Freezing point on Centigrade scale is +0 degrees; boiling point, 100 degrees.

Freezing point on Reaumur scale is +0 degrees; boiling point, 80 degrees.

Of water at sea level at normal barometer pressure (29.9 inch).

The "absolute zero" of temperature denotes that condition of matter at which heat ceases to exist. At this point a body would be wholly deprived of heat and a gas would exert no pressure.

The absolute zero on the Fahrenheit scale is about 461 degrees below zero.

The absolute zero on the Centigrade scale is about 274 degrees below zero.

The absolute zero on the Reaumur scale is about 219 degrees below zero.

An English unit of heat (B. T. U.) is the quantity required to raise one pound of water one degree Fahrenheit. A metric unit of heat or metric caloric (M. C.) is the quantity of heat required to raise one litre of water one degree Centigrade.

Table of Decimal Equivalents of 8ths, 16ths, 32nds, and 64ths of an Inch

8ths

$\frac{1}{8}$ = .125

$\frac{1}{4}$ = .250

$\frac{3}{8}$ = .375

$\frac{1}{2}$ = .500

$\frac{5}{8}$ = .625

$\frac{3}{4}$ = .750

$\frac{7}{8}$ = .875

16ths

$\frac{1}{16}$ = .0625

$\frac{3}{16}$ = .1875

$\frac{5}{16}$ = .3125

$\frac{7}{16}$ = .4375

$\frac{9}{16}$ = .5625

$\frac{11}{16}$ = .6875

$\frac{13}{16}$ = .8125

$\frac{15}{16}$ = .9375

32nds

$\frac{1}{32}$ = .03125

$\frac{3}{32}$ = .09375

$\frac{5}{32}$ = .15625

$\frac{7}{32}$ = .21875

$\frac{9}{32}$ = .28125

$\frac{11}{32}$ = .34375

$\frac{13}{32}$ = .40625

$\frac{15}{32}$ = .46875

$\frac{17}{32}$ = .53125

$\frac{19}{32}$ = .59375

$\frac{21}{32}$ = .65625

$\frac{23}{32}$ = .71875

$\frac{25}{32}$ = .78125

$\frac{27}{32}$ = .84375

$\frac{29}{32}$ = .90625

$\frac{31}{32}$ = .96875

64ths

$\frac{1}{64}$ = .015625

$\frac{3}{64}$ = .046875

$\frac{5}{64}$ = .078125

$\frac{7}{64}$ = .109375

$\frac{9}{64}$ = .140625

$\frac{11}{64}$ = .171875

$\frac{13}{64}$ = .203125

$\frac{15}{64}$ = .234375

$\frac{17}{64}$ = .265625

$\frac{19}{64}$ = .296875

$\frac{21}{64}$ = .328125

$\frac{23}{64}$ = .359375

$\frac{25}{64}$ = .390625

$\frac{27}{64}$ = .421875

$\frac{29}{64}$ = .453125

$\frac{31}{64}$ = .484375

$\frac{33}{64}$ = .515625

$\frac{35}{64}$ = .546875

$\frac{37}{64}$ = .578125

$\frac{39}{64}$ = .609375

$\frac{41}{64}$ = .640625

$\frac{43}{64}$ = .671875

$\frac{45}{64}$ = .703125

$\frac{47}{64}$ = .734375

$\frac{49}{64}$ = .765625

$\frac{51}{64}$ = .796875

$\frac{53}{64}$ = .828125

$\frac{55}{64}$ = .859375

$\frac{57}{64}$ = .890625

$\frac{59}{64}$ = .921875

$\frac{61}{64}$ = .953125

$\frac{63}{64}$ = .984375

Decimal Equivalents of Millimeters and Fractions of Millimeters

$\frac{1}{100}$ mm. = .0003937″

mm.	Inches	mm.	Inches	mm.	Inches	mm.	Inches
$\frac{1}{50}$	= .00079	$\frac{39}{50}$	= .03071	27	= 1.06299	64	= 2.51968
$\frac{2}{50}$	= .00157	$\frac{40}{50}$	= .03150	28	= 1.10236	65	= 2.55905
$\frac{3}{50}$	= .00236	$\frac{41}{50}$	= .03228	29	= 1.14173	66	= 2.59842
$\frac{4}{50}$	= .00315	$\frac{42}{50}$	= .03307				
		$\frac{43}{50}$	= .03386	30	= 1.18110	67	= 2.63779
$\frac{5}{50}$	= .00394	$\frac{44}{50}$	= .03465	31	= 1.22047	68	= 2.67716
$\frac{6}{50}$	= .00472			32	= 1.25984	69	= 2.71653
$\frac{7}{50}$	= .00551	$\frac{45}{50}$	= .03543	33	= 1.29921	70	= 2.75590
$\frac{8}{50}$	= .00630	$\frac{46}{50}$	= .03622	34	= 1.33858	71	= 2.79527
$\frac{9}{50}$	= .00709	$\frac{47}{50}$	= .03701				
		$\frac{48}{50}$	= .03780	35	= 1.37795	72	= 2.83464
$\frac{1}{32}$	= .00787	$\frac{49}{50}$	= .03858	36	= 1.41732	73	= 2.87401
$\frac{1}{32}$	= .00866			37	= 1.45669	74	= 2.91338
$\frac{1}{32}$	= .00945	1	= .03937	38	= 1.49606	75	= 2.95275
$\frac{1}{32}$	= .01024	2	= .07874	39	= 1.53543	76	= 2.99212
		3	= .11811				
$\frac{1}{32}$	= .01102	4	= .15748	40	= 1.57480	77	= 3.03149
$\frac{1}{32}$	= .01181			41	= 1.61417	78	= 3.07086
$\frac{1}{32}$	= .01260	5	= .19685	42	= 1.65354	79	= 3.11023
$\frac{1}{32}$	= .01339	6	= .23622	43	= 1.69291	80	= 3.14960
$\frac{1}{32}$	= .01417	7	= .27559	44	= 1.73228	81	= 3.18897
$\frac{1}{32}$	= .01496	8	= .31496				
		9	= .35433	45	= 1.77165	82	= 3.22834
$\frac{1}{32}$	= .01575			46	= 1.81102	83	= 3.26771
$\frac{1}{32}$	= .01654	10	= .39370	47	= 1.85039	84	= 3.30708
$\frac{1}{32}$	= .01732	11	= .43307	48	= 1.88976	85	= 3.34645
$\frac{1}{32}$	= .01811	12	= .47244	49	= 1.92913	86	= 3.38582
$\frac{1}{32}$	= .01890	13	= .51181				
		14	= .55118	50	= 1.96850	87	= 3.42519
$\frac{1}{32}$	= .01969			51	= 2.00787	88	= 3.46456
$\frac{1}{32}$	= .02047	15	= .59055	52	= 2.04724	89	= 3.50393
$\frac{1}{32}$	= .02126	16	= .62992	53	= 2.08661	90	= 3.54330
$\frac{1}{32}$	= .02205	17	= .66929	54	= 2.12598	91	= 3.58267
$\frac{1}{32}$	= .02283	18	= .70866				
$\frac{1}{32}$	= .02362	19	= .74803	55	= 2.16535	92	= 3.62204
$\frac{1}{32}$	= .02441			56	= 2.20472	93	= 3.66141
$\frac{1}{32}$	= .02520	20	= .78740	57	= 2.24409	94	= 3.70078
$\frac{1}{32}$	= .02598	21	= .82677	58	= 2.28346	95	= 3.74015
$\frac{1}{32}$	= .02677	22	= .86614	59	= 2.32283	96	= 3.77952
		23	= .90551				
$\frac{1}{32}$	= .02756	24	= .94488	60	= 2.36220	97	= 3.81889
$\frac{1}{32}$	= .02835			61	= 2.40157	98	= 3.85826
$\frac{1}{32}$	= .02913	25	= .98425	62	= 2.44094	99	= 3.89763
$\frac{1}{32}$	= .02992	26	= 1.02362	63	= 2.48031	100	= 3.93700

10 mm. = 1 Centimeter = 0.3937 inches.
10 cm. = 1 Decimeter = 3.937 inches.
10 dm. = 1 Meter = 39.37 Inches.
25.4 mm. = 1 English Inch.

Circumferences and Areas of Circles

Advancing by Eighths

Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area
1/64	.04909	.00019	2 5/8	8.2467	5.4119	6 3/4	21.206	35.785
1/32	.09818	.00077	2 11/16	8.4430	5.6727	6 7/8	21.598	37.122
3/64	.14726	.00173	2 3/4	8.6394	5.9396			
1/16	.19635	.00307	2 13/16	8.8357	6.2126	7	21.991	38.485
3/32	.29452	.00690	2 7/8	9.0321	6.4918	7 1/8	22.384	39.871
1/8	.39270	.01227	2 15/16	9.2284	6.7771	7 1/4	22.776	41.282
5/32	.49087	.01917				7 3/8	23.169	42.718
3/16	.58905	.02761	3	9.4248	7.0686	7 1/2	23.562	44.179
7/32	.68722	.03758	3 1/16	9.6211	7.3662	7 5/8	23.955	45.664
			3 1/8	9.8175	7.6699	7 3/4	24.347	47.173
1/4	.78540	.04909	3 3/16	10.014	7.9798	7 7/8	24.740	48.707
9/32	.88357	.06213	3 1/4	10.210	8.2958			
5/16	.98175	.07670	3 5/16	10.407	8.6179	8	25.133	50.265
11/32	1.0799	.09281	3 3/8	10.603	8.9462	8 1/8	25.525	51.849
3/8	1.1781	.11045	3 7/16	10.799	9.2806	8 1/4	25.918	53.456
13/32	1.2763	.12962	3 1/2	10.996	9.6211	8 3/8	26.311	55.088
7/16	1.3744	.15033	3 9/16	11.192	9.9678	8 1/2	26.704	56.745
15/32	1.4726	.17257	3 5/8	11.388	10.321	8 5/8	27.096	58.426
			3 11/16	11.585	10.680	8 3/4	27.489	60.132
1/2	1.5708	.19635	3 3/4	11.781	11.045	8 7/8	27.882	61.862
17/32	1.6690	.22166	3 13/16	11.977	11.416			
9/16	1.7671	.24850	3 7/8	12.174	11.793	9	28.274	63.617
19/32	1.8653	.27688	3 15/16	12.370	12.177	9 1/8	28.667	65.397
5/8	1.9635	.30680				9 1/4	29.060	67.201
21/32	2.0617	.33824	4	12.566	12.566	9 3/8	29.452	69.029
11/16	2.1598	.37122	4 1/16	12.763	12.962	9 1/2	29.845	70.882
23/32	2.2580	.40574	4 1/8	12.959	13.364	9 5/8	30.238	72.760
			4 3/16	13.155	13.772	9 3/4	30.631	74.662
3/4	2.3562	.44179	4 1/4	13.352	14.186	9 7/8	31.023	76.589
25/32	2.4544	.47937	4 5/16	13.548	14.607			
13/16	2.5525	.51849	4 3/8	13.744	15.033	10	31.416	78.540
27/32	2.6507	.55914	4 7/16	13.941	15.466	10 1/8	31.809	80.516
7/8	2.7489	.60132	4 1/2	14.137	15.904	10 1/4	32.201	82.516
29/32	2.8471	.64504	4 9/16	14.334	16.349	10 3/8	32.594	84.541
15/16	2.9452	.69029	4 5/8	14.530	16.800	10 1/2	32.987	86.590
31/32	3.0434	.73708	4 11/16	14.726	17.257	10 5/8	33.379	88.664
			4 3/4	14.923	17.721	10 3/4	33.772	90.763
1	3.1416	.7854	4 13/16	15.119	18.190	10 7/8	34.165	92.886
1 1/16	3.3379	.8866	4 7/8	15.315	18.665			
1 1/8	3.5343	.9940	4 15/16	15.512	19.147	11	34.558	95.033
1 3/16	3.7306	1.1075				11 1/8	34.950	97.205
1 1/4	3.9270	1.2272	5	15.708	19.635	11 1/4	35.343	99.402
1 5/16	4.1233	1.3530	5 1/16	15.904	20.129	11 3/8	35.736	101.62
1 3/8	4.3197	1.4849	5 1/8	16.101	20.629	11 1/2	36.128	103.87
1 7/16	4.5160	1.6230	5 3/16	16.297	21.135	11 5/8	36.521	106.14
1 1/2	4.7124	1.7671	5 1/4	16.493	21.648	11 3/4	36.914	108.43
1 9/16	4.9087	1.9175	5 5/16	16.690	22.166	11 7/8	37.306	110.75
1 5/8	5.1051	2.0739	5 3/8	16.886	22.691			
1 11/16	5.3014	2.2365	5 7/16	17.082	23.221	12	37.699	113.10
1 3/4	5.4978	2.4053	5 1/2	17.279	23.758	12 1/8	38.092	115.47
1 13/16	5.6941	2.5802	5 9/16	17.475	24.301	12 1/4	38.485	117.86
1 7/8	5.8905	2.7612	5 5/8	17.671	24.850	12 3/8	38.877	120.28
1 15/16	6.0868	2.9483	5 11/16	17.686	25.406	12 1/2	39.270	122.72
			5 3/4	18.064	25.967	12 5/8	39.663	125.19
2	6.2832	3.1416	5 13/16	18.261	26.535	12 3/4	40.055	127.68
2 1/16	6.4795	3.3410	5 7/8	18.457	27.109	12 7/8	40.448	130.19
2 1/8	6.6759	3.5466	5 15/16	18.653	27.688			
2 3/16	6.8722	3.7583				13	40.841	132.73
2 1/4	7.0686	3.9761	6	18.850	28.274	13 1/8	41.233	135.30
2 5/16	7.2649	4.2000	6 1/8	19.242	29.465	13 1/4	41.626	137.89
2 3/8	7.4613	4.4301	6 1/4	19.635	30.680	13 3/8	42.019	140.50
2 7/16	7.6576	4.6664	6 3/8	20.028	31.919	13 1/2	42.412	143.14
2 1/2	7.8540	4.9087	6 1/2	20.420	33.183	13 5/8	42.804	145.80
2 9/16	8.0503	5.1572	6 5/8	20.813	34.472	13 3/4	43.197	148.49
						13 7/8	43.590	151.29

Circumferences and Areas of Circles—Continued

Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area
14	43.982	153.94	21 3/8	67.152	358.84	28 3/4	90.321	649.18
14 1/8	44.375	156.70	21 1/2	67.544	363.05	28 7/8	90.713	654.84
14 1/4	44.768	159.48	21 5/8	67.937	367.28			
14 3/8	45.160	162.30	21 3/4	68.330	371.54	29	91.106	660.52
14 1/2	45.553	165.13	21 7/8	68.722	375.83	29 1/8	91.499	666.23
14 5/8	45.946	167.99				29 1/4	91.892	671.96
14 3/4	46.338	170.87	22	69.115	380.13	29 3/8	92.284	677.71
14 7/8	46.731	173.78	22 1/8	69.508	384.46	29 1/2	92.677	683.49
			22 1/4	69.900	388.82	29 5/8	93.070	689.30
15	47.124	176.71	22 3/8	70.293	393.20	29 3/4	93.462	695.13
15 1/8	47.517	179.67	22 1/2	70.686	397.61	29 7/8	93.855	700.98
15 1/4	47.909	182.65	22 5/8	71.079	402.04			
15 3/8	48.302	185.66	22 3/4	71.471	406.49	30	94.248	706.86
15 1/2	48.695	188.69	22 7/8	71.864	410.97	30 1/8	94.640	712.70
15 5/8	49.087	191.75				30 1/4	95.033	718.69
15 3/4	49.480	194.83	23	72.257	415.48	30 3/8	95.426	724.64
15 7/8	49.873	197.93	23 1/8	72.649	420.00	30 1/2	95.819	730.62
			23 1/4	73.042	424.56	30 5/8	96.211	736.62
16	50.265	201.06	23 3/8	73.435	429.13	30 3/4	96.604	742.64
16 1/8	50.658	204.22	23 1/2	73.827	433.74	30 7/8	96.997	748.69
16 1/4	51.051	207.39	23 5/8	74.220	438.36			
16 3/8	51.444	210.60	23 3/4	74.613	443.01	31	97.389	754.77
16 1/2	51.836	213.82	23 7/8	75.006	447.69	31 1/8	97.782	760.87
16 5/8	52.229	217.08				31 1/4	98.175	766.99
16 3/4	52.622	220.35	24	75.398	452.39	31 3/8	98.567	773.14
16 7/8	53.014	223.65	24 1/8	75.791	457.11	31 1/2	98.960	779.31
			24 1/4	76.184	461.86	31 5/8	99.353	785.51
17	53.407	226.98	24 3/8	76.576	466.64	31 3/4	99.746	791.73
17 1/8	53.800	230.33	24 1/2	76.969	471.44	31 7/8	100.138	797.98
17 1/4	54.192	233.71	24 5/8	77.362	476.26			
17 3/8	54.585	237.10	24 3/4	77.754	481.11	32	100.531	804.25
17 1/2	54.978	240.53	24 7/8	78.147	485.98	32 1/8	100.924	810.54
17 5/8	55.371	243.98				32 1/4	101.316	816.86
17 3/4	55.763	247.45	25	78.540	490.87	32 3/8	101.709	823.21
17 7/8	56.156	250.95	25 1/8	78.933	495.79	32 1/2	102.102	829.58
			25 1/4	79.325	500.74	32 5/8	102.494	835.97
18	56.549	254.47	25 3/8	79.718	505.71	32 3/4	102.887	842.39
18 1/8	56.941	258.02	25 1/2	80.111	510.71	32 7/8	103.280	848.83
18 1/4	57.334	261.59	25 5/8	80.503	515.72			
18 3/8	57.727	265.18	25 3/4	80.896	520.77	33	103.673	855.30
18 1/2	58.119	268.80	25 7/8	81.289	525.84	33 1/8	104.065	861.79
18 5/8	58.512	272.45				33 1/4	104.458	868.31
18 3/4	58.905	276.12	26	81.681	530.93	33 3/8	104.851	874.85
18 7/8	59.298	279.81	26 1/8	82.074	536.05	33 1/2	105.243	881.41
			26 1/4	82.467	541.19	33 5/8	105.636	888.00
19	59.690	283.53	26 3/8	82.860	546.35	33 3/4	106.029	894.62
19 1/8	60.083	287.27	26 1/2	83.252	551.55	33 7/8	106.421	901.26
19 1/4	60.476	291.04	26 5/8	83.645	556.76			
19 3/8	60.868	294.83	26 3/4	84.038	562.00	34	106.814	907.92
19 1/2	61.261	298.65	26 7/8	84.430	567.27	34 1/8	107.207	914.61
19 5/8	61.654	302.49				34 1/4	107.600	921.32
19 3/4	62.046	306.35	27	84.823	572.56	34 3/8	107.992	928.06
19 7/8	62.439	310.24	27 1/8	85.216	577.87	34 1/2	108.385	934.82
			27 1/4	85.608	583.21	34 5/8	108.778	941.61
20	62.832	314.16	27 3/8	86.001	588.57	34 3/4	109.170	948.42
20 1/8	63.225	318.10	27 1/2	86.394	593.96	34 7/8	109.563	955.25
20 1/4	63.617	322.06	27 5/8	86.786	599.37			
20 3/8	64.010	326.05	27 3/4	87.179	604.81	35	109.956	962.11
20 1/2	64.403	330.06	27 7/8	87.572	610.27	35 1/8	110.348	969.00
20 5/8	64.795	334.10				35 1/4	110.741	975.91
20 3/4	65.188	338.16	28	87.965	615.75	35 3/8	111.134	982.84
20 7/8	65.581	342.25	28 1/8	88.357	621.26	35 1/2	111.527	989.80
			28 1/4	88.750	626.80	35 5/8	111.919	996.87
21	65.973	346.36	28 3/8	89.143	632.36	35 3/4	112.312	1003.8
21 1/8	66.366	350.50	28 1/2	89.535	637.94	35 7/8	112.705	1010.8
21 1/4	66.759	354.66	28 5/8	89.928	643.55			

Circumferences and Areas of Circles—Continued

Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area
36	113.097	1017.9	43¼	135.874	1469.1	50½	158.650	2003.0
36⅛	113.490	1025.0	43⅜	136.267	1477.6	50⅝	159.043	2012.9
36¼	113.883	1032.1	43½	136.659	1486.2	50¾	159.436	2022.8
36⅜	114.275	1039.2	43⅝	137.052	1491.7	50⅞	159.829	2032.8
36½	114.668	1046.3	43¾	137.445	1503.3			
36⅝	115.061	1053.5	43⅞	137.837	1511.9	51	160.221	2042.8
36¾	115.454	1060.7				51⅛	160.614	2052.8
36⅞	115.846	1068.0	44	138.230	1520.5	51¼	161.007	2062.9
			44⅛	138.623	1529.2	51⅜	161.399	2073.0
37	116.239	1075.2	44¼	139.015	1537.9	51½	161.792	2083.1
37⅛	116.632	1082.5	44⅜	139.408	1546.6	51⅝	162.185	2093.2
37¼	117.024	1089.8	44½	139.801	1555.3	51¾	162.577	2103.3
37⅜	117.417	1097.1	44⅝	140.194	1564.0	51⅞	162.970	2113.5
37½	117.810	1104.5	44¾	140.586	1572.8			
37⅝	118.202	1111.8	44⅞	140.979	1581.6	52	163.363	2123.7
37¾	118.596	1119.2				52⅛	163.756	2133.9
37⅞	118.988	1126.7	45	141.372	1590.4	52¼	164.148	2144.2
			45⅛	141.764	1599.3	52⅜	164.541	2154.5
38	119.381	1134.1	45¼	142.157	1608.2	52½	164.934	2164.8
38⅛	119.773	1141.0	45⅜	142.550	1617.0	52⅝	165.326	2175.1
38¼	120.166	1149.1	45½	142.942	1626.0	52¾	165.719	2185.4
38⅜	120.559	1156.6	45⅝	143.335	1634.9	52⅞	166.112	2195.8
38½	120.951	1164.2	45¾	143.728	1643.9			
38⅝	121.344	1171.7	45⅞	144.121	1652.9	53	166.504	2206.2
38¾	121.737	1179.3				53⅛	166.897	2216.6
38⅞	122.129	1186.9	46	144.513	1661.9	53¼	167.290	2227.0
			46⅛	144.906	1670.9	53⅜	167.683	2237.5
39	122.522	1194.6	46¼	145.299	1680.0	53½	168.075	2248.0
39⅛	122.915	1202.3	46⅜	145.691	1689.1	53⅝	168.468	2258.5
39¼	123.308	1210.0	46½	146.084	1698.2	53¾	168.861	2269.1
39⅜	123.700	1217.7	46⅝	146.477	1707.4	53⅞	169.253	2279.6
39½	124.093	1225.4	46¾	146.869	1716.5			
39⅝	124.486	1233.2	46⅞	147.262	1725.7	54	169.646	2290.2
39¾	124.878	1241.0				54⅛	170.039	2300.8
39⅞	125.271	1248.8	47	147.655	1734.9	54¼	170.431	2311.5
			47⅛	148.048	1744.2	54⅜	170.824	2322.1
40	125.664	1256.6	47¼	148.440	1753.5	54½	171.217	2332.8
40⅛	126.056	1264.5	47⅜	148.338	1762.7	54⅝	171.609	2343.5
40¼	126.449	1272.4	47½	149.226	1772.1	54¾	172.002	2354.3
40⅜	126.842	1280.3	47⅝	149.618	1781.4	54⅞	172.395	2365.0
40½	127.235	1288.2	47¾	150.011	1790.8			
40⅝	127.627	1296.2	47⅞	150.404	1800.1	55	172.788	2375.8
40¾	128.020	1304.2				55⅛	173.180	2386.6
40⅞	128.413	1312.2	48	150.796	1809.6	55¼	173.573	2397.5
			48⅛	151.189	1819.0	55⅜	173.966	2408.3
41	128.805	1320.3	48¼	151.582	1828.5	55½	174.358	2419.2
41⅛	129.198	1328.3	48⅜	151.975	1837.9	55⅝	174.751	2430.1
41¼	129.591	1336.4	48½	152.367	1847.5	55¾	175.144	2441.1
41⅜	129.983	1344.5	48⅝	152.760	1857.0	55⅞	175.536	2452.0
41½	130.376	1352.7	48¾	153.153	1866.5			
41⅝	130.769	1360.8	48⅞	153.545	1876.1	56	175.929	2463.0
41¾	131.161	1369.0				56⅛	176.322	2474.0
41⅞	131.554	1377.2	49	153.938	1885.7	56¼	176.715	2485.0
			49⅛	154.331	1895.4	56⅜	177.107	2496.1
42	131.947	1385.4	49¼	154.723	1905.0	56½	177.500	2507.2
42⅛	132.340	1393.7	49⅜	155.116	1914.7	56⅝	177.893	2518.3
42¼	132.732	1402.0	49½	155.509	1924.4	56¾	178.285	2529.4
42⅜	133.125	1410.3	49⅝	155.902	1934.2	56⅞	178.678	2540.6
42½	133.518	1418.6	49¾	156.294	1943.9			
42⅝	133.910	1427.0	49⅞	156.687	1953.7	57	179.071	2551.8
42¾	134.303	1435.4				57⅛	179.463	2563.0
42⅞	134.696	1443.8	50	157.080	1963.5	57¼	179.856	2574.2
			50⅛	157.472	1973.3	57⅜	180.249	2585.4
43	135.088	1452.2	50¼	157.865	1983.2	57½	180.642	2596.7
43⅛	135.481	1460.7	50⅜	158.258	1993.1	57⅝	181.034	2608.0

Circumferences and Areas of Circles—Continued

Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area
57¾	181.427	2619.4	65	204.204	3318.3	72¼	226.980	4099.8
57⅞	181.820	2630.7	65⅛	204.596	3331.1	72⅜	227.373	4114.0
			65¼	204.989	3343.9	72½	227.765	4128.2
58	182.212	2642.1	65⅜	205.382	3356.7	72⅝	228.158	4142.5
58⅛	182.605	2653.5	65½	205.774	3369.6	72¾	228.551	4156.8
58¼	182.998	2664.9	65⅝	206.167	3382.4	72⅞	228.994	4171.1
58⅜	183.390	2676.4	65¾	206.560	3395.3			
58½	183.783	2687.8	65⅞	206.952	3408.2	73	229.336	4185.4
58⅝	184.176	2699.3				73⅛	229.729	4199.7
58¾	184.569	2710.9	66	207.345	3421.2	73¼	230.122	4214.1
58⅞	184.961	2722.4	66⅛	207.738	3434.2	73⅜	230.514	4228.5
			66¼	208.131	3447.2	73½	230.907	4242.9
59	185.354	2734.0	66⅜	208.523	3460.2	73⅝	231.300	4257.4
59⅛	185.747	2745.6	66½	208.916	3473.2	73¾	231.692	4271.8
59¼	186.139	2757.2	66⅝	209.309	3486.3	73⅞	232.085	4286.3
59⅜	186.532	2768.8	66¾	209.701	3499.4			
59½	186.925	2780.5	66⅞	210.094	3512.5	74	232.478	4300.8
59⅝	187.317	2792.2				74⅛	232.871	4315.4
59¾	187.710	2803.9	67	210.487	3525.7	74¼	233.263	4329.9
59⅞	188.103	2815.7	67⅛	210.879	3538.8	74⅜	233.656	4344.5
			67¼	211.272	3552.0	74½	234.049	4359.2
60	188.496	2827.4	67⅜	211.665	3565.2	74⅝	234.441	4373.8
60⅛	188.888	2839.2	67½	212.058	3578.5	74¾	234.834	4388.5
60¼	189.281	2851.0	67⅝	212.450	3591.7	74⅞	235.227	4403.1
60⅜	189.674	2862.9	67¾	212.843	3605.0			
60½	190.006	2874.8	67⅞	213.236	3618.3	75	235.619	4417.9
60⅝	190.459	2886.6				75⅛	236.012	4432.6
60¾	190.852	2898.6	68	213.628	3631.7	75¼	236.405	4447.4
60⅞	191.244	2910.5	68⅛	214.021	3645.0	75⅜	236.798	4462.2
			68¼	214.414	3658.4	75½	237.190	4477.0
61	191.637	2922.5	68⅜	214.806	3671.8	75⅝	237.583	4491.8
61⅛	192.030	2934.5	68½	215.199	3685.3	75¾	237.976	4506.7
61¼	192.423	2946.5	68⅝	215.592	3698.7	75⅞	238.368	4521.5
61⅜	192.815	2958.5	68¾	125.984	3712.2			
61½	193.208	2970.6	68⅞	216.377	3725.7	76	238.761	4536.5
61⅝	193.601	2982.7				76⅛	239.154	4551.4
61¾	193.993	2994.8	69	216.770	3739.3	76¼	239.546	4566.4
61⅞	194.386	3006.9	69⅛	217.163	3752.8	76⅜	239.939	4581.3
			69¼	217.555	3766.4	76½	240.332	4596.3
62	194.779	3019.1	69⅜	217.948	3780.0	76⅝	240.725	4611.4
62⅛	195.171	2031.3	69½	218.341	3793.7	76¾	241.117	4626.4
62¼	195.564	3043.5	69⅝	218.733	3807.3	76⅞	241.510	4641.5
62⅜	195.957	3055.7	68¾	219.126	3821.0			
62½	196.350	3068.0	69⅞	219.519	3834.7	77	241.903	4656.6
62⅝	196.742	3080.3				77⅛	242.295	4671.8
62¾	197.135	3092.6	70	219.911	3848.5	77¼	242.688	4686.9
62⅞	197.528	3104.9	70⅛	220.304	3862.2	77⅜	243.081	4702.1
			70¼	220.697	3876.0	77½	243.473	4717.3
63	197.920	3117.2	70⅜	221.090	3889.8	77⅝	243.866	4732.5
63⅛	198.313	3129.6	70½	221.482	3903.6	77¾	244.259	4747.8
63¼	198.706	3142.0	70⅝	221.875	3917.5	77⅞	244.652	4763.1
63⅜	199.098	3154.5	70¾	222.268	3931.4			
63½	199.491	3166.9	70⅞	222.660	3945.3	78	245.044	4778.4
63⅝	199.884	3179.4				78⅛	245.437	4793.7
63¾	200.277	3191.9	71	223.053	3959.2	78¼	245.830	4809.0
63⅞	200.669	3204.4	71⅛	223.446	3973.1	78⅜	246.222	4824.4
			71¼	223.838	3987.1	78½	246.615	4839.8
64	201.062	3217.0	71⅜	224.231	4001.1	78⅝	247.008	4855.2
64⅛	201.455	3229.6	71½	224.624	4015.2	78¾	247.400	4870.7
64¼	201.847	3242.2	71⅝	225.017	4029.2	78⅞	247.793	4886.2
64⅜	202.240	3254.8	71¾	225.409	4043.3			
64½	202.633	3267.5	71⅞	225.802	4057.4	79	248.186	4901.7
64⅝	203.025	3280.1				79⅛	248.579	4917.2
64¾	203.418	3292.8	72	226.195	4071.5	79¼	248.971	4932.7
64⅞	203.811	3305.6	72⅛	226.587	4085.7	79⅜	249.364	4948.3

Circumferences and Areas of Circles—Continued

Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area
79 1/8	249.757	4963.9	86 3/8	271.355	5859.6	93 1/4	292.954	6829.5
79 5/8	250.149	4979.5	86 1/2	271.748	5876.5	93 3/8	293.346	6847.8
79 3/4	250.542	4995.2	86 5/8	272.140	5893.5	93 1/2	293.739	6866.1
79 7/8	250.935	5010.9	86 3/4	272.533	5910.6	93 5/8	294.132	6884.5
			86 7/8	272.926	5927.6	93 3/4	294.524	6902.9
80	251.327	5026.5				93 7/8	294.917	6921.3
80 1/8	251.720	5042.3	87	273.319	5944.7			
80 1/4	252.113	5058.0	87 1/8	273.711	5961.8	94	295.310	6939.8
80 3/8	252.506	5073.8	87 1/4	274.104	5978.9	94 1/8	295.702	6958.2
80 1/2	252.898	5089.6	87 3/8	274.497	5996.0	94 1/4	296.095	6976.7
80 5/8	253.291	5105.4	87 1/2	274.889	6103.2	94 3/8	296.488	6995.3
80 3/4	253.684	5121.2	87 5/8	275.282	6030.4	94 1/2	296.881	7013.8
80 7/8	254.076	5137.1	87 3/4	275.675	6047.6	94 5/8	297.273	7032.4
			87 7/8	276.067	6064.9	94 3/4	297.666	7051.0
81	254.469	5153.0				94 7/8	298.059	7069.6
81 1/8	254.862	5168.9	88	276.460	6082.1			
81 1/4	255.254	5184.9	88 1/8	276.853	6099.4	95	298.451	7088.2
81 3/8	255.647	5200.8	88 1/4	277.246	6116.7	95 1/8	298.844	7106.9
81 1/2	256.040	5216.8	88 3/8	277.638	6134.1	95 1/4	299.237	7125.6
81 5/8	256.433	5232.8	88 1/2	278.031	6151.4	95 3/8	299.629	7144.3
81 3/4	256.825	5248.9	88 5/8	278.424	6168.8	95 1/2	300.022	7163.0
81 7/8	257.218	5264.9	88 3/4	278.816	6186.2	95 5/8	300.415	7181.8
			88 7/8	279.209	6203.7	95 3/4	300.807	7200.6
82	257.611	5281.0				95 7/8	301.200	7219.4
82 1/8	258.003	5297.1	89	279.602	6221.1			
82 1/4	258.396	5313.3	89 1/8	279.994	6238.6	96	301.593	7238.2
82 3/8	258.789	5329.4	89 1/4	280.387	6256.1	96 1/8	301.986	7257.1
82 1/2	259.181	5345.6	89 3/8	280.780	6273.7	96 1/4	302.378	7276.0
82 5/8	259.574	5361.8	89 1/2	281.173	6291.2	96 3/8	302.771	7294.9
82 3/4	259.967	5378.1	89 5/8	281.565	6308.8	96 1/2	303.164	7313.8
82 7/8	260.359	5394.3	89 3/4	281.958	6326.4	96 5/8	303.556	7332.8
			89 7/8	282.351	6344.1	96 3/4	303.949	7351.8
83	260.752	5410.6				96 7/8	304.342	7370.8
83 1/8	261.145	5426.9	90	282.743	6361.7			
83 1/4	261.538	5443.3	90 1/8	283.136	6379.4	97	304.734	7389.8
83 3/8	261.930	5459.6	90 1/4	283.529	6397.1	97 1/8	305.127	7408.9
83 1/2	262.323	5476.0	90 3/8	283.921	6441.9	97 1/4	305.520	7428.0
83 5/8	262.716	5492.4	90 1/2	284.314	6432.6	97 3/8	305.913	7447.1
83 3/4	263.108	5508 8	90 5/8	284.707	6450.4	97 1/2	306.305	7466.2
83 7/8	263.501	5525.3	90 3/4	285.100	6468.2	97 5/8	306.698	7485.3
			90 7/8	285.492	6486.0	97 3/4	307.091	7504.5
84	263.894	5541 8				97 7/8	307.483	7523.7
84 1/8	264.286	5558.3	91	285.885	6503.9			
84 1/4	264.679	5574.8	91 1/8	286.278	6521.8	98	307.876	7543.0
84 3/8	265.072	5591.4	91 1/4	286.670	6539.7	98 1/8	308.269	7562.2
84 1/2	255.465	5607.9	91 3/8	287.063	6557.6	98 1/4	308.661	7581.5
84 5/8	265.857	5624.5	91 1/2	287.456	6575.5	98 3/8	309.054	7600.8
84 3/4	266.250	5641.2	91 5/8	287.848	6593.5	98 1/2	309.447	7620.1
84 7/8	266.643	5657.8	91 3/4	288.241	6611.5	98 5/8	309.840	7639.5
			91 7/8	288.634	6629.6	98 3/4	310.232	7658.9
85	267.035	5674.5				98 7/8	310.625	7678.3
85 1/8	267.428	5691.2	92	289.027	6647.6			
85 1/4	267.821	5707.9	92 1/8	289.419	6665.7	99	311.018	7697.7
85 3/8	268.213	5724.7	92 1/4	289.812	6683.8	99 1/8	311.410	7717.1
85 1/2	268.606	5741.5	92 3/8	290.205	6701.9	99 1/4	311.803	7736.6
85 5/8	268.999	5758.3	92 1/2	290.597	6720.1	99 3/8	312.196	7756.1
85 3/4	269.392	5775.1	92 5/8	290.990	6738.2	99 1/2	312.588	7775.6
85 7/8	269.784	5791.9	92 3/4	291.383	6756.4	99 5/8	312.981	7795.2
			92 7/8	291.775	6774.7	99 3/4	313.374	7814.8
86	270.177	5808.8				99 7/8	313.767	7834.4
86 1/8	270.570	5825.7	93	292.168	6792.9			
86 1/4	270.962	5842.6	93 1/8	292.561	6811.2	100	314.159	7854.0

Coefficients of Linear Expansion at Temperatures between 32° and 222° Fahr.

Material	For 1° Cent.	For 1° Fahr.	Material	For 1° Cent.	For 1° Fahr.
Aluminum—cast	.0000222	.0000123	Steel—untempered	.0000108	.0000060
Aluminum—rolled	.0000207	.0000115	Steel—tempered	.0000126	.0000070
Antimony	.0000110	.0000061	Tin	.0000207	.0000115
Bismuth	.0000139	.0000077	Zinc	.0000288	.0000160
Brass	.0000189	.0000105	Brick—best stock	.0000055	.0000031
Copper	.0000171	.0000095	Fire Brick	.0000049	.0000027
Gold	.0000153	.0000085	Building { From.	.0000072	.0000040
Iron—Cast	.0000108	.0000060	Stones { To	.0000144	.0000080
Iron—Wrought	.0000117	.0000065	Glass	.0000088	.0000049
Lead	.0000284	.0000158	Porcelain	.0000036	.0000020
Nickel	.0000126	.0000070	Roman Cement		
Platinum	.0000087	.0000048	dry	.0000144	.0000080
Silver	.0000198	.0000110	Slate	.0000104	.0000058
			Wedgewood ware	.0000088	.0000049

Melting Points or Temperatures of Fusion

Solid	Cent.	Fahr.	Solid	Cent.	Fahr.
Aluminum	656	1214	Magnesium	750	1382
Antimony	630	1166	Manganese	1207	2205
Bismuth	268	514	Mercury	—39.7	—39.5
Brass	1030	1886	Nickel	1435	2615
Bronze	920	1688	Palladium	1546	2815
Cadmium	320	608	Platinum	1753	3187
Chromium	1487	2709	Potassium	62	144
Cobalt	1463	2665	Rhodium	2000	3632
Copper	1084	1983	Silver	953	1747
Gold	1064	1947	Sodium	95	203
Iridium	2500	4532	Steel—mild	1475	2687
Iron—cast, gray	1220 to 1530	2228 to 2786	Steel—hard	1420	2588
Iron—cast, white	1050 to 1135	1922 to 2075	Tin	232	449
Iron—wrought	1500 to 1600	2732 to 2912	Vanadium	1775	3227
Lead	327	620	Zinc	419	786

Mensuration of Surfaces and Volumes

Area of rectangle = length × breadth.
Area of triangle = base × ½ perpendicular height.
Diameter of circle = radius × 2.
Circumference of circle = diameter × 3.1416
Area of circle = square of diameter × .7854.
Area of sector of circle = $\frac{\text{area of circle} \times \text{number of degrees in arc.}}{360}$
Area of surface of cylinder = circumference × length + area of two ends.
To find the diameter of circle having given area: Divide the area by .7854, and extract the square root.
To find the volume of a cylinder: Multiply the area of the section in square inches by the length in inches = the volume in cubic inches. Cubic inches divided by 1728 = volume in cubic feet.
Surface of a sphere = square of diameter × 3.1416
Solidity of a sphere = cube of diameter × .5236.
Side of an inscribed cube = radius of a sphere × 1.1547.
Area of the base of a pyramid or cone, whether round, square or triangular, multiplied by one-third of its height = the solidity.
Diam. × .8862 = side of an equal square.
Diam. × .7071 = side of an inscribed square.
Radius × 6.2832 = circumference.
Circumference = 3.5446 × √ Area of circle.
Diameter = 1.1283 × √ Area of circle.
Length of arc = No. of degrees × .017453 radius.
Degrees in arc whose length equals radius = 57° 2958′.
Length of an arc of 1° = radius × .017543.
Length of an arc of 1 Min. = radius × .0002909.
Length of an arc of 1 Sec. = radius × .0000048.

	π =	Proportion of circumference to diameter = 3.1415926.
	π^2 =	9.8696044.
	$\sqrt{\pi}$ =	1.7724538.
Log.	π =	0.49715.
	$1/\pi$ =	0.31831.
	1/360 =	.002778.
	$360/\pi$ =	114.59.

Mensuration of Surfaces and Volumes—Continued

Lineal feet	×	.00019	=	Miles
" yards	×	.0006	=	"
Square inches	×	.007	=	Square feet.
" feet	×	.111	=	" yards
" yards	×	.0002067	=	Acres.
Acres	×	4840.	=	Square yards.
Cubic inches	×	.00058	=	Cubic feet
" feet	×	.03704	=	" yards
Circular inches	×	.00546	=	Square feet
Cyl. inches	×	.0004546	=	Cubic feet
" feet	×	.02909	=	" yards
Links	×	.22	=	Yards
"	×	.66	=	Feet
Feet	×	1.5	=	Links
Width in chains	×	8.	=	Acres per mile
183346 circular inches			=	1 square foot
2200 cylindrical inches			=	1 cubic foot
Cubic feet	×	7.48	=	U. S. gallons
" inches	×	.004329	=	" "
U. S. Gallons	×	.13367	=	Cubic feet
" "	×	231.	=	" inches
Cubic feet	×	.8036	=	U. S. bushel
" inches	×	.000468	=	" "
Cyl. feet of water	×	6.	=	U. S. gallons
Lbs. Avoir.	×	.009	=	Cwt. (112)
" "	×	.00045	=	Tons (2240)
Cubic feet of water	×	62.5	=	Lbs. Avoir.
" inch of water	×	.03617	=	" "
Cyl. feet water	×	49.1	=	" "
" inch of water	×	.02842	=	" "
12 U. S. gallons of water			=	1 cwt.
240 U. S. gallons of water			=	1 ton
1.8 cubic feet of water			=	1 cwt.
35.88 cubic feet of water			=	1 ton
Column of water, 12 inches high, and 1 inch in diameter			=	.341 Lbs.
U. S. bushel	×	.0495	=	Cubic yards.
" "	×	1.2446	=	" feet
" "	×	2150.42	=	Cubic Inches

Comparative Table of the United States and Metric Systems

Denomination	Equivalent
One grain equals in grammes	0.0648
One pound avoirdupois equals in kilogrammes	0.4536
One ton of 2240 pounds equals in tonnes	1.0160
One ton of 2000 pounds equals in tonnes	0.9071
One inch equals in millimetres	25.400
One foot equals in metres	0.3048
One mile equals in kilometres	1.6094
One square inch equals in square millimetres	645.2
One square foot equals in square metres	0.09291
One acre equals in ares (100 square metres)	40.47
One square mile equals in square kilometres	2.590
One cubic inch equals in cubic centimetres	16.39
One cubic foot equals in cubic metres	0.02832
One cubic yard equals in cubic metres	0.7646
One quart dry measure equals in litres	1.101
One quart liquid or wine measure equals in litres	0.9465
One foot pound equals in kilogrammetres	0.1383
One pound per foot equals in kilogrammes per metre	1.488
One thousand pounds per square inch equals in kilogrammes per square millimetres	0.703
One pound per square foot equals in kilogrammes per square metre	4.882
One pound per cubic foot equals in kilogrammes per cubic metre	16.02
One degree Fahrenheit equals in degrees Centigrade.	0.5556

Comparative Table of the United States and Metric Systems

Denomination	Equivalent
One gramme equals in grains	15.433
One kilogramme equals in pounds avoirdupois	2.2047
One tonne equals in tons of 2240 pounds	0.9843
One tonne equals in tons of 2000 pounds	1.1024
One millimetre equals in inches	0.0394
One metre equals in feet	3.2807
One kilometre equals in miles	0.6213
One square millimetre equals in square inches	0.00155

Comparative Table of United States and Metric Systems—Continued

Denomination	Equivalent
One square metre equals in square feet	10.763
One are (100 square metres) equals in acres	0.02471
One square kilometre equals in square miles	0.3861
One cubic centimetre equals in cubic inches	0.0610
One cubic metre or stere equals in cubic feet	35.3105
One cubic metre equals in cubic yards	1.3078
One litre (one cubic decimetre) equals in cubic inches	61.017
One litre equals in quarts, dry measure	0.908
One litre equals in quarts, liquid or wine measure	1.0566
One kilogrammetre equals in foot pounds	7.2313
One kilogramme per metre equals in pounds per foot	0.6720
One kilogramme per square millimetre equals in pounds per square inch	1422.
One kilogramme per square metre equals in pounds per square foot	0.2048
One kilogramme per cubic metre equals in pounds per cubic foot	0.0624
One degree Centigrade equals in degrees Fahrenheit.	1.8

Metric Conversion Table

Millimetres × .03937 = inches.
Millimetres ÷ 25.4 = inches.
Centimetres × .3937 = inches.
Centimetres ÷ 2.54 = inches.
Metres × 39.37 = inches.
Metres × 3.281 = feet.
Metres × 1.094 = yards.
Kilometres × .621 = miles
Kilometres ÷ 1.6093 = miles.
Kilometres × 3280.8693 = feet.
Sq. Millimetres × .00155 = sq. in.
Sq. Millimetres ÷ 645.1 = sq. in.
Sq. Centimetres × .155 = sq. in.
Sq. Centimetres ÷ 6.451 = sq. in.
Sq. Metres × 10.764 = sq. ft.
Sq. Kilometres × 247.1 = acres.
Hectare × 2.471 = acres.
Cu. Centimetres ÷ 16.383 = cu. in.
Cu. Centimetres ÷ 3.69 = fl. drams.
Cu. Centimetres ÷ 29.57 = fluid oz.
Cu. Metres × 35.315 = cu. ft.
Cu. Metres × 1.308 = cu. yds.
Cu. Metres × 264.2 = gals. (231 cu. in.)
Litres × 61.022 = cu. in.
Litres × 33.84 = fluid oz.
Litres × .2642 = gals. (231 cu. in.)
Litres ÷ 3.78 = gals. (231 cu. in.)
Litres ÷ 28.316 = cu. ft.
Hectolitres × 3.531 = cu. ft.
Hectolitres × 2.84 = Bu. (2150.42 cu. in.)
Hectolitres × .131 = cu. yds.
Hectolitres ÷ 26.42 = gals. (231 cu.in.)
Grammes × 15.432 = grains.
Grammes ÷ 981 = dynes.
Grammes (water) ÷ 29.57 = fluid oz.
Grammes ÷ 28.35 = oz. avoirdupois
Grammes per cu. cent. ÷ 27.7 = lbs. per cu. in.
Joule × .7373 = ft. lbs.
Kilo-grammes × 2.2046 = pounds.
Kilo-grammes × 35.3 = oz. avoirdupois.
Kilo-grammes ÷ 907.2 = tons (2000 lbs.)
Kilo-grammes per sq. cent. × 14.223 = lbs. per sq. in.
Kilo-gram-metres × 7.233 = ft. lbs.
Kilo-gr. per Metre × .672 = lbs. per ft.
Kilo-gr. per cu. Metre × .062 = lbs. per cu. ft.
Kilo-gr. per Cheval × 2.235 = lbs. per H. P.
Kilo-Watts × 1.34 = Horsepower
Watts ÷ 746. = Horsepower.
Watts × .7373 = ft. pounds p. second
Calorie × 3.968 = B. T. U.
Cheval vapeur × .9863 = Horsepower
(Centigrade × 1.8) + 32 = deg. Fahr.
Franc × .193 = Dollars
Gravity Paris = 980.94 centimetres per sec.

Calorific Power, Carbon Value, and Evaporative Power of Various Fuels.

TOTAL HEAT OF COMBUSTION, OR CALORIFIC POWER OF A FUEL.—The calorific power of a fuel is the number of units of heat produced by the combustion of 1 pound weight of it. The unit of heat is the amount of heat required to raise 1 pound of water 1° Fahr.

CARBON VALUE OF FUEL.—The carbon value of any fuel is the weight of carbon in pounds having the same calorific value as 1 pound of the fuel. Carbon value equals calorific power of fuel divided by calorific power of carbon.

THEORETICAL EVAPORATIVE POWER OF FUEL.—The theoretical evaporative power of fuel is stated in pounds of water evaporated from and at 212° Fahr., and is obtained by dividing the calorific power of the fuel by 966.

ACTUAL EVAPORATIVE POWER OF COAL IN STEAM BOILERS. From numerous experiments on steam boilers, it appears that the actual evaporative power of coal varies from 50 per cent to 85 per cent of the theoretical evaporative power. An average of a considerable number of tests gave the actual evaporative power equal to 70 per cent of the theoretical evaporative power of the coal.

Combustible		Calorific Power in British Thermal Units	Carbon Value	Evaporative Power in Lbs. of Water from and at 212° Fahr.
Carbon—burned to carbonic acid		14544	1.000	15.06
Carbon—burned to carbonic oxide		4451	.306	4.61
Carbonic oxide		4325	.297	4.48
Marsh gas		23513	1.617	24.34
Olefiant gas		21344	1.468	22.10
Hydrogen		62032	4.265	64.22
Hydrogen, deducting latent heat in steam formed		53338	3.667	55.22
Sulphur		3996	.275	4.14
Straw—with 16 per cent water		5200	.358	5.38
Wood—kiln dried		8000	.550	8.28
Wood—air dried, with 20 per cent water		5600	.385	5.80
Peat—kiln dried		10000	.688	10.35
Peat—air dried, with 20 per cent water		6500	.447	6.73
Charcoal from wood—dry		13000	.894	13.46
Charcoal from peat—dry		11600	.798	12.01
Coal—lignite—air dried		11000	.756	11.39
Coal—bituminous	from	13000	.894	13.46
	to	15700	1.079	16.25
	average	14100	.969	14.60
Coal—anthracite	from	14000	.963	14.49
	to	16200	1.114	16.77
	average	15000	1.031	15.53
Coke	from	12000	.825	12.42
	to	13700	.942	14.18
Block fuel		15000	1.031	15.53
Petroleum		20000	1.375	20.70
Natural gas (Pennsylvania)		26000	1.788	26.92

Weight and Specific Gravity of Metals (Kent)

	Specific Gravity Range According to Several Authorities	Specific Gravity Approximate Mean Value Used in Calculation of Weight	Weight per Cubic Foot Pounds	Weight per Cubic Inch Pounds
Aluminum	2.56 to 2.71	2.67	166.5	.0963
Antimony	6.66 to 6.86	6.76	421.6	.2439
Bismuth	9.74 to 9.90	9.82	612.4	.3544
Brass, copper and zinc				
80 20		8.60	536.3	.3103
70 30	7.8 to 8.6	8.40	523.8	.3031
60 40		8.36	521.3	.3017
50 50		8.20	511.4	.2959
Bronze { copper 95 to 80 / tin 5 to 20 }	8.52 to 8.96	8.853	552.0	.3195
Cadmium	8.6 to 8.7	8.65	539.0	.3121
Calcium	1.58			
Chromium	5.0			
Cobalt	8.5 to 8.6			
Gold, pure	19.245 to 19.361	19.258	1200.9	.6949
Copper	8.69 to 8.92	8.853	552.0	.3195
Iridium	22.38 to 23.0		1396.0	.8076
Iron, cast	6.85 to 7.48	7.218	450.0	.2604
Iron, wrought	7.4 to 7.9	7.70	480.0	.2779
Lead	11.07 to 11.44	11.38	709.7	.4106
Manganese	7.0 to 8.0	8.00	499.0	.2887
Magnesium	1.69 to 1.75	1.75	109.0	.0641
Mercury 32°	13.60 to 13.62	13.62	849.3	.4915
Mercury 60°	13.58	13.58	846.8	.4900
Mercury 212°	13.37 to 13.38	13.38	834.4	.4828
Nickel	8.279 to 8.93	8.8	548.7	.3175
Platinum	20.33 to 22.07	21.5	1347.0	.7758
Potassium	0.865			
Silver	10.474 to 10.511	10.505	655.1	.3791
Sodium	0.97			
Steel	7.69* to 7.932†	7.854	489.6	.2834
Tin	7.291 to 7.409	7.350	458.3	.2652
Titanium	5.3			
Tungsten	17.0 to 17.6			
Zinc	6.86 to 7.20	7.00	436.5	.2526

*Hard and burned. †Very pure and soft. The specific gravity decreases as the carbon is increased. In the first column of figures, the lowest are usually those of cast metals, which are more or less porous; the highest are of metals finely rolled or drawn into wire.

Proportions of Various Compositions in Common Use

(In One Hundred Parts)

Babbit's metal.....Tin 89, copper 3.7, antimony 7.3
Fine yellow brass.....Copper 66, zinc 34
Gun metal, valves, etc.....Copper 90, tin 10
White brass.....Copper 10, zinc, 80, tin 10
German silver.....Copper 33.3, zinc 33.4, nickel 33.3
Church bells.....Copper 80, zinc 5.6, tin 10.1, lead 4.3
Gongs.....Copper 81.6, tin 18.4
Lathe bushes.....Copper 80, tin 20
Machinery bearings.....Copper 87.5, tin 12.5
Muntz metal.....Copper 60, zinc 40
Sheathing metal.....Copper 56, zinc 44

Various Tables Showing Weights of Materials

Weight Per Bushel of Different Grains, Etc.

Barley	48 pounds	Flax Seed	56 pounds
Beans	63 pounds	Hemp Seed	48 pounds
Buckwheat	46 pounds	Oats	32 pounds
Blue Grass Seed	14 pounds	Peas	64 pounds
Corn	56 pounds	Rye	56 pounds
Corn Meal	50 pounds	Salt	80 pounds
Clover Seed	60 pounds	Timothy Seed	45 pounds
Dried Apples	22 pounds	Wheat	60 pounds
Dried Peaches	33 pounds	Potatoes (heaped)	60 pounds

Weight Per Barrel of, Different Articles

Flour	196 pounds	Fish	200 pounds
Salt	280 pounds	Soap	256 pounds
Beef	200 pounds	Cement	300 pounds
Pork	200 pounds		

56 pounds of butter equals	1 firkin
100 pounds of meal or flour equals	1 sack
100 pounds of grain or flour equals	1 central
100 pounds of dry fish equals	1 quintal
100 pounds of nails equals	1 cask

Miscellaneous Articles

One ton of (2240 pounds) cured hay equals	425 cubic feet
One ton of hay in mow equals	414.37 cubic feet, or a cube of 7½ ft
Hay, as usually delivered	5 pounds per cubic foot
Hay, well pressed	8 pounds per cubic foot
Straw, loose	3½ pounds per cubic foot
Straw, well pressed	5⅓ pounds per cubic foot
One gallon of water (U. S.)	8.33 pounds
One gallon of oil	7¾ pounds
One gallon of molasses	11⅔ pounds
One gallon of alcohol	6.9 pounds
One gallon of spirits of turpentine	7.31 pounds
One keg of powder	25 pounds

Weights, in Pounds, of Various Articles

As rated by Railway Companies, when their weights cannot otherwise be ascertained

Ashes, pot or pearl	barrel,	450	pounds
Apples and barreled fruits	barrel,	200	pounds
Apples	bushel,	50	pounds
Barley	bushel,	45	pounds
Beef, pork, bacon	per hhd.,	1000	pounds
Butter, tallow, lard	per bbl.,	333	pounds
Salt fish and meat	per firkin,	100	pounds
Bran, feed, shipstuffs, oats	bushel,	35	pounds
Buckwheat	bushel,	48	pounds
Bricks, common	each	5	pounds
Bark	cord,	2000	pounds
Charcoal	bushel,	22	pounds
Coke and cake meal	bushel,	40	pounds
Clover seed	bushel,	62	pounds
Eggs	barrel,	200	pounds
Fish and salt meat	per firkin,	100	pounds
Flour and meal	per bushel, 56 pounds; barrel,	216	pounds
Grain and seeds, not stated	bushel,	60	pounds
Hides, green	each,	85	pounds
Hides, dry, salted or spanish	each,	33	pounds
Ice, coal, lime	bushel,	80	pounds
Liquors, malt and distilled	barrel,	30	pounds

Weights, in Pounds, of Various Articles—Continued

Liquors	per gallon,	10 pounds
Lumber—pine, poplar, hemlock	foot B. M.,	4 pounds
Lumber—oak, walnut, cherry, ash	foot, B. M.,	5 pounds
Nails and spikes	keg,	106 pounds
Onions, wheat, potatoes	bushel,	60 pounds
Oysters	per bushel, 100 pounds; per 1000,	350 pounds
Plastering lath	per 1000,	600 pounds
Rosin, tar, turpentine	barrel,	300 pounds
Sand, gravel, etc	per cubic foot,	150 pounds
Shingles	per 1000, short 900 pounds, long,	1400 pounds
Salt	bushel	70 pounds
Stone, undressed	perch,	4000 pounds
Stone, dressed	per cubic foot,	180 pounds
Timothy and light grass seed	bushel,	40 pounds
Wood—hickory	cord,	4500 pounds
Wood—oak	cord,	3500 pounds

WEIGHT OF ONE CUBIC FOOT OF PURE WATER

At 32° F. (freezing point)	62.418 pounds
At 39.1° F. (maximum density)	62.425 pounds
At 62° F. (standard temperature)	62.355 pounds
At 212° F. (boiling point, under 1 atmosphere)	59.76 pounds
American gallon equals 231 cubic inches of water at 62° F. equals	8.3356 pounds
British gallon equals 277.274 cubic inches of water at 62° F. equals	10 pounds

Weight and Specific Gravity of Liquids

Liquids at 32° F.	Weight of one Cubic Foot Pounds	Weight of one Gallon (Imperial) Pounds	Specific Gravity Water = 1
Mercury	848.7	136.0	13.596
Bromine	185.1	29.7	2.966
Sulphuric acid	114.9	18.4	1.84
Nitrous acid	96.8	15.5	1.55
Chloroform	95.5	15.3	1.53
Water of the Dead Sea	77.4	12.4	1.24
Nitric acid	76.2	12.2	1.22
Acetic acid	67.4	10.8	1.08
Milk	64.3	10.3	1.03
Sea water	64.05	10.3	1.026
Pure water (distilled) at 39° F	62.425	10.0	1.0
Wine of Bordeaux	62.9	9.9	0.994
Wine of Burgandy	61.9	9.9	0.991
Oil, linseed	58.7	9.4	0.94
Oil, poppy	58.1	9.3	0.93
Oil, rape seed	57.4	9.2	0.92
Oil, whale	57.4	9.2	0.92
Oil, olive	57.1	9.15	0.159
Oil, turpentine	54.3	8.7	0.87
Oil, potato	51.2	8.2	0.82
Petroleum	54.9	8.8	0.88
Naptha	53.1	8.5	0.85
Ether, nitric	69.3	11.1	1.11
Ether, sulphurous	67.4	10.8	1.08
Ether, nitrous	55.6	8.9	0.89
Ether, acetic	55.6	8.9	0.89
Ether, hydrochloric	54.3	8.7	0.87
Ether, sulphuric	44.9	7.2	0.72
Alcohol, proof spirit	57.4	9.2	0.92
Alcohol, pure	49.3	7.9	0.79
Benzine	53.1	8.5	0.85
Wood Spirit	49.9	8.0	0.80

Circular and Angular Measure

60 seconds (″) = 1 minute (′)
60 minutes = 1 degree (°)
360 degrees = 1 circumference (C)

Cubic Measure

1728 cubic inches = 1 cubic, or solid foot.
27 cubic feet = 1 cubic, or solid yard.

A pile of wood cut 4 feet long, piled 4 feet high, 8 feet long = 128 cubic feet = 1 cord.

A perch of stone = 16½ feet long, by 1 foot high, by 1½ feet thick = 24¾ cubic feet.

A perch of stone = 22 cubic feet in Philadelphia.

A perch of stone = 16½ cubic feet in some New England States.

The perch is so variable in different localities that it should never be used in making a contract unless the contents in cubic feet be specified.

A ton (2240 pounds) of Pennsylvania anthracite, when broken for domestic use, occupies from 41 to 43 cubic feet of space, the mean of which is equal to 1.556 cubic yards, or a cube of 3.476 feet on each edge.

A ton (2240 pounds) of bituminous coal equals 44 to 48 cubic feet, mean equal to 1.704 cubic yards; or a cube of 3.583 feet on each edge.

A ton (2240 pounds) of coke = 80 cubic feet.

A cubic foot is equal to ..... 1728 cubic inches
A cubic foot is equal to ..... 0.037037 cubic yards
A cubic foot is equal to .. 0.803564 U. S. struck bushel of 2150.42 cubic inches
A cubic foot is equal to ..... 3.21426 U. S. pecks
A cubic foot is equal to ..... 7.48052 U. S. liquid gallons of 231 cubic inches
A cubic foot is equal to ... 6.42851 U. S. dry gallons of 268.8025 cubic inches
A cubic foot is equal to ..... 29.9220[illegible] U. S. liquid quarts
A cubic foot is equal to ..... 25.71405 U. S. dry quarts
A cubic foot is equal to ..... 59.84416 U. S. liquid pints
A cubic foot is equal to ..... 51.42809 U. S. dry pints
A cubic foot is equal to ..... 239.37662 U. S. gills
A cubic foot is equal to ..... 0.26667 flour barrel of 3 struck bushels
A cubic foot is equal to ..... 0.23748 U. S. liquid barrel of 31½ gallons

A cubic yard is equal to 7.2 flour barrels of 3 struck bushels each.

A ton in computing the tonnage of a ship or other vessel is 100 cubic feet of their internal space.

A ton in computing freight on ships is taken at 40 cubic feet or 2240 pounds, at the ship's option.

Dry Measure

	Edge of a cube of equal capacity
2 pints = 1 quart	4.066 inches
4 quarts = 1 gallon = 8 pints	6.454 inches
2 gallons = 1 peck = 16 pints	8.131 inches
4 pecks = 1 bushel (struck) = 64 pints = 32 quarts = 8 gallons	12.908 inches

A gallon dry measure = 268.8 cubic inches.

A bushel dry meaure (same as British Winchester struck bushel) = 2150.42 cubic inches, or 77.63 pounds avoirdupois of pure water at its maximum density.

The dimensions of a bushel by law are 18½ inches inner diameter, 19½ inches outer diameter, and 8 inches deep; and when heaped, the cone is not to be less than 6 inches high, which makes a heaped bushel equal to 1¼ struck bushels, or to 1.56 cubic feet.

A struck bushel = 1.24 cubic feet.

The dry flour barrel = 3.75 cubic feet = 3 struck bushels. The dry barrel is not however, a legalized measure.

36 heaped bushels = 1 chaldron.

Measures of Length

12 inches = 1 foot.
3 feet = 1 yard = 36 inches.
5½ yards = 1 rod = 198 inches = 16½ feet.
80 rods = 1 furlong = 7920 inches = 660 feet = 220 yards.
4 furlongs = 1 mile = 63,360 inches = 5280 feet = 1760 yards = 320 rods.

Gunter's Chain

(Sometimes used in Land Surveying)
7.92 inches = 1 link.
100 links = 1 chain = 4 rods = 66 feet.
80 chains = 1 mile.

Ropes and Cables

6 feet = 1 fathom; 120 fathoms = 1 cable's length.

The United States standard yard is the same as the imperial yard of Great Britain. It is determined as follows; The rod of a pendulum vibrating seconds of mean time in the latitude of London in a vacuum at the level of the sea is divided into 391,393 equal parts, and 360,000 of these parts are 36 inches, or 1 standard yard.

An inch is one 500,500,000th part of the earth's polar axis.

Artificers sometimes divide the inch into lines or twelfths, but more commonly into binary divisions—half, quarter, eighth, sixteenth and thirty-second.

Mechanical engineers divide the inch decimally—10ths, 100ths, 1000ths, etc.

Civil engineers divide the foot decimally.

A nautical mile, geographical mile, sea mile, or knot, as adopted by United States Coast and Geodetic Survey, is equal to 6080.27 feet.

British Admiralty knot = 6080 feet.

A geographical or nautical mile may be taken = 1.15 statute miles.

The league = 3 nautical miles.

The geographical degree = 60 geographical or nautical miles.

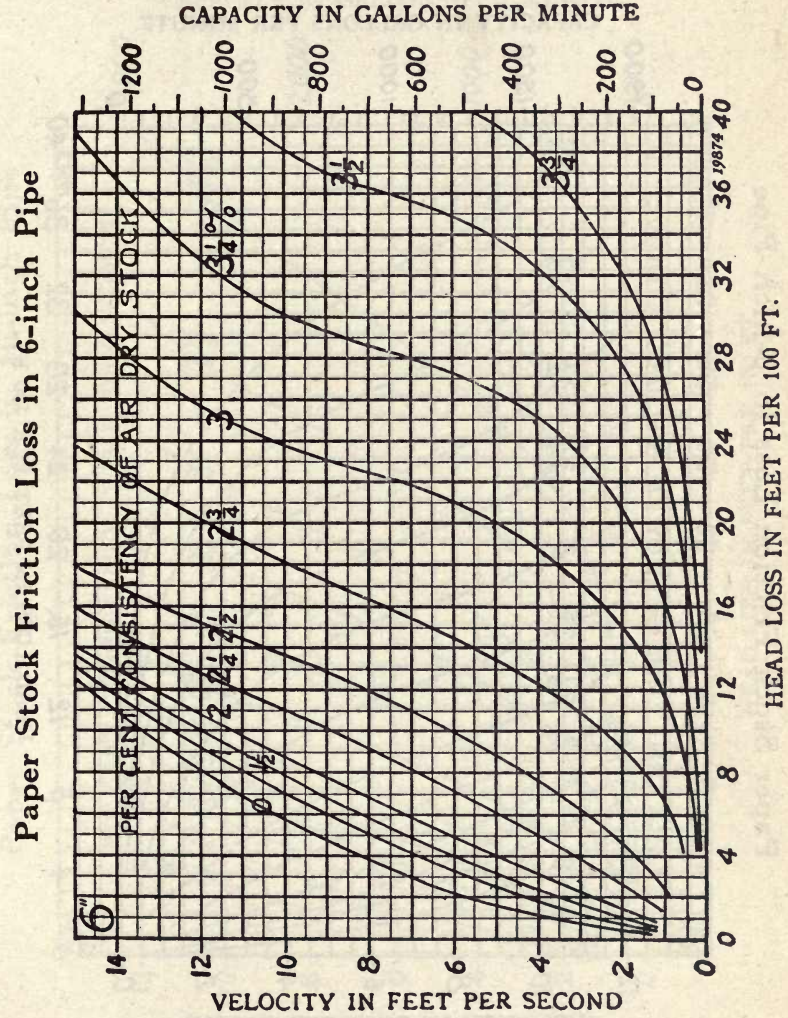
The length of a degree of latitude varies, being 68.72 miles at the equator, 69.05 miles in middle latitudes, and 69.34 miles in the polar regions. A degree of longitude is greatest at the equator, where it is 69.16 miles, and it gradually decreases toward the poles, where it is 0.

1 hand = 4 inches.
1 pace = 3 feet.

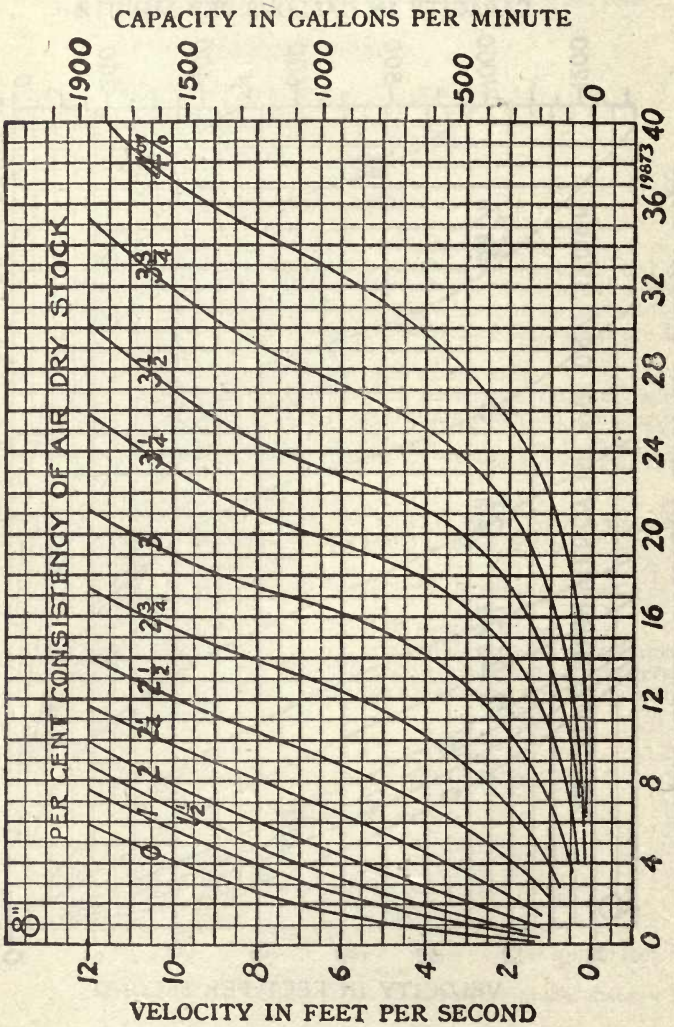
The hand is used for heights of horses and girths of spars.

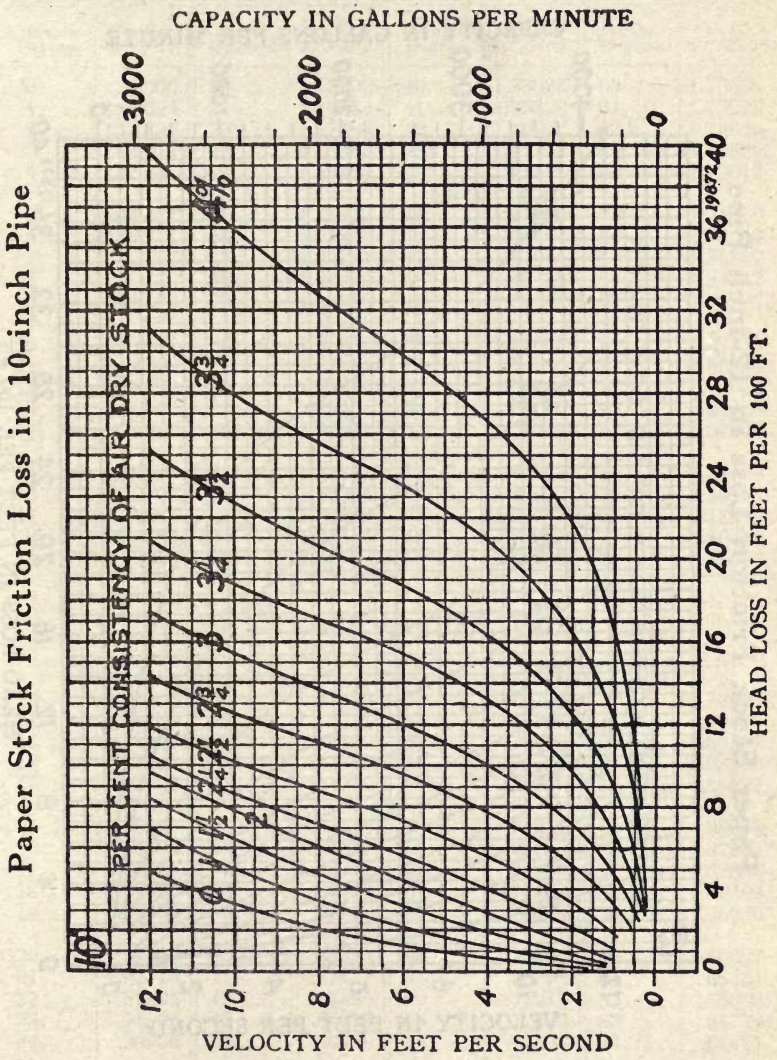
Square or Land Measure

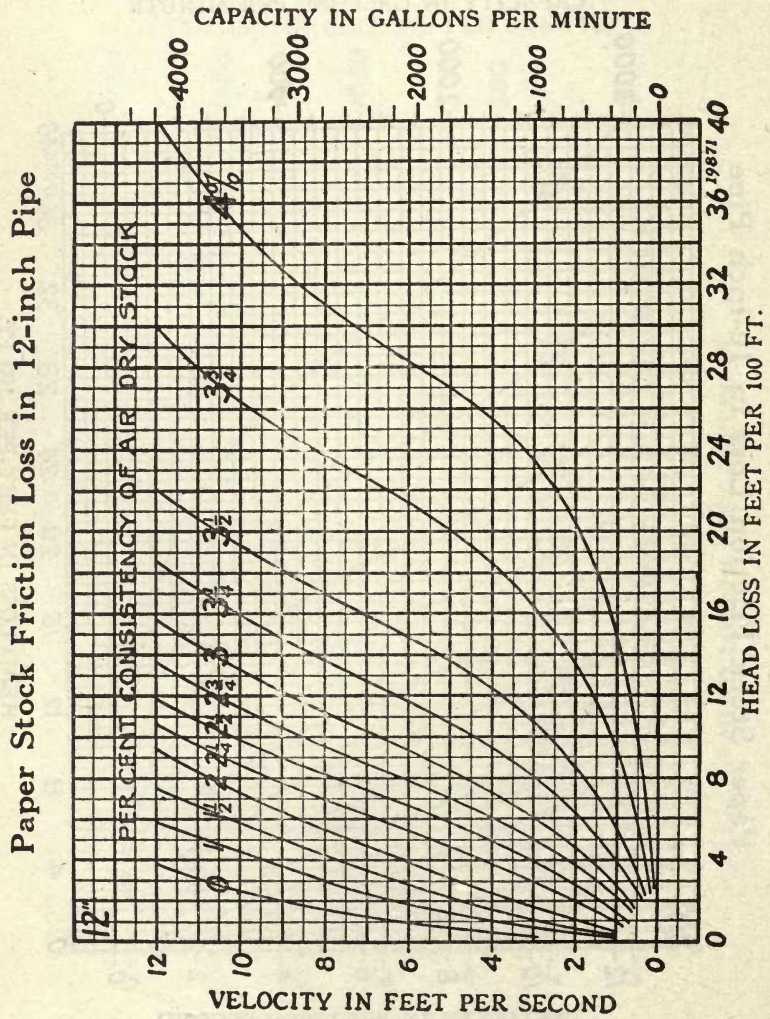
144 square inches = 1 square foot.
9 square feet = 1 square yard = 1296 square inches.
30¼ square yards = 1 square rod = 272¼ square feet.
40 square rods = 1 rood = 1210 square yards = 10,890 square feet.
4 roods = 1 acre = 160 square rods = 1840 square yards = 43,560 square feet.
A section of land = 640 acres = 1 square mile.
208.71 feet square = 43.560 square feet = 10 square Gunter's chains = 1 acre, or 220 x 198 feet = 1 acre.
A square ½-acre is 147.58 feet at each side; or 110x198 feet.
A square ¼- acre is 104.355 feet at each side; or 55 x 198 feet.
A circular acre is 235.504 feet in diameter.
A circular ½-acre is 166.527 feet in diameter.
A circular ¼-acre is 117.752 feet in diameter.
A circular inch is a circle of 1-inch diameter; a square foot = 183.346 circular inches.
1 square inch = 1.27324 circular inches; and 1 circular inch = 0.7854 of a square inch.


Paper Stock Friction Loss in 6-inch Pipe
CAPACITY IN GALLONS PER MINUTE
1200
1000
800
600
400
200
0
PER CENT CONSISTENCY OF AIR DRY STOCK
6"
½
¾
1
1¼
1½
2
2¼
2½
2¾
3
3½
3¾
4
VELOCITY IN FEET PER SECOND
14
12
10
8
6
4
2
0
HEAD LOSS IN FEET PER 100 FT.
40
36
32
28
24
20
16
12
8
4
0
1987A

Paper Stock Friction Loss in 8-inch Pipe




Paper Stock Friction Loss in 10-inch Pipe
CAPACITY IN GALLONS PER MINUTE
3000
2000
1000
0
PER CENT CONSISTENCY OF AIR DRY STOCK
HEAD LOSS IN FEET PER 100 FT.
0
4
8
12
16
20
24
28
32
36
40
VELOCITY IN FEET PER SECOND
12
10
8
6
4
2
0


Paper Stock Friction Loss in 12-inch Pipe
CAPACITY IN GALLONS PER MINUTE
4000
3000
2000
1000
0
PER CENT CONSISTENCY OF AIR DRY STOCK
12"
0
1/2
1
1 1/2
2
2 1/2
3
3 1/2
4
4 1/2
5
6
7
8
9
10
HEAD LOSS IN FEET PER 100 FT.
0
4
8
12
16
20
24
28
32
36
40
VELOCITY IN FEET PER SECOND
12
10
8
6
4
2
0

United States Standard Baume Scale

Relation Between Baumé Degrees and Specific Gravity

LIQUIDS HEAVIER THAN WATER

Baume Degrees	Sp. Gr. 60°/60°F	Baume Degrees	Sp. Gr. 60°/60°F	Baume Degrees	Sp. Gr. 60°/60°F	Baume Degrees	Sp. Gr. 60°/60°F
0	1.00000	20	1.16000	40	1.38095	60	1.70588
1	1.00694	21	1.16935	41	1.39423	61	1.72619
2	1.01399	22	1.17886	42	1.40777	62	1.74699
3	1.02113	23	1.18852	43	1.42157	63	1.76829
4	1.02837	24	1.19835	44	1.43564	64	1.79012
5	1.03571	25	1.20833	45	1.45000	65	1.81250
6	1.04317	26	1.21849	46	1.46465	66	1.83544
7	1.05072	27	1.22881	47	1.47959	67	1.85897
8	1.05839	28	1.23932	48	1.49485	68	1.88312
9	1.06618	29	1.25000	49	1.51042	69	1.90789
10	1.07407	30	1.26087	50	1.52632	70	1.93333
11	1.08209	31	1.27193	51	1.54255	71	1.95946
12	1.09023	32	1.28319	52	1.55914	72	1.98630
13	1.09848	33	1.29464	53	1.57609	73	2.01389
14	1.10687	34	1.30631	54	1.59341	74	2.04225
15	1.11538	35	1.31818	55	1.61111	75	2.07143
16	1.12403	36	1.33028	56	1.62921	76	2.10145
17	1.13281	37	1.34259	57	1.64773	77	2.13235
18	1.14173	38	1.35514	58	1.66667	78	2.16418
19	1.15079	39	1.36792	59	1.68605	79	2.19697

LIQUIDS LIGHTER THAN WATER

10	1.00000	30	0.87500	50	0.77778	70	0.70000
11	.99291	31	.86957	51	.77348	71	.69652
12	.98592	32	.86420	52	.76923	72	.69307
13	.97902	33	.85890	53	.76503	73	.68966
14	.97222	34	.85366	54	.76087	74	.68627
15	.96552	35	.84848	55	.75676	75	.68293
16	.95890	36	.84337	56	.75269	76	.67961
17	.95238	37	.83832	57	.74866	77	.67633
18	.94595	38	.83333	58	.74468	78	.67308
19	.93960	39	.82840	59	.74074	79	.66986
20	.93333	40	.82353	60	.73684	80	.66667
21	.92715	41	.81871	61	.73298	81	.66351
22	.92105	42	.81395	62	.72917	82	.66038
23	.91503	43	.80925	63	.72539	83	.65728
24	.90909	44	.80460	64	.72165	84	.65421
25	.90323	45	.80000	65	.71795	85	.65117
26	.89744	46	.79545	66	.71428	86	.64815
27	.89172	47	.79096	67	.71066	87	.64516
28	.88608	48	.78652	68	.70707	88	.64220
29	.88050	49	.78212	69	.70352	89	.63927

From Circular No. 59 Bureau of Standards.

Table of Degrees Brix

Per Cent. Sugar (Degrees Balling's or Brix) with Corresponding Specific Gravity and Degrees Baume. Temperature 60° F.

Per Cent Sugar Balling's or Brix 60° F— 15.56° C.	Specific Gravity 60°/60° F	Degrees Baume 60° F	Per Cent Sugar Balling's or Brix 60° F— 15.56° C	Specific Gravity 60°/60° F	Degrees Baume 60° F	Per Cent Sugar Balling's or Brix 60° F— 15.56° C	Specific Gravity 60°/60° F	Degrees Baume 60° F
0	1.0000	0.00	34	1.1491	18.81	68	1.3384	36.67
1	1.0039	0.56	35	1.1541	19.36	69	1.3447	37.17
2	1.0078	1.13	36	1.1591	19.90	70	1.3509	37.66
3	1.0118	1.68	37	1.1641	20.44	71	1.3573	38.17
4	1.0157	2.24	38	1.1692	20.98	72	1.3636	38.66
5	1.0197	2.80	39	1.1743	21.52	73	1.3700	39.16
6	1.0238	3.37	40	1.1794	22.06	74	1.3764	39.65
7	1.0278	3.93	41	1.1846	22.60	75	1.3829	40.15
8	1.0319	4.49	42	1.1898	23.13	76	1.3894	40.64
9	1.0360	5.04	43	1.1950	23.66	77	1.3959	41.12
10	1.0402	5.60	44	1.2003	24.20	78	1.4025	41.61
11	1.0443	6.15	45	1.2057	24.74	79	1.4091	42.10
12	1.0485	6.71	46	1.2110	25.26	80	1.4157	42.58
13	1.0528	7.28	47	1.2164	25.80	81	1.4224	43.06
14	1.0570	7.81	48	1.2218	26.32	82	1.4291	43.54
15	1.0613	8.38	49	1.2273	26.86	83	1.4359	44.02
16	1.0657	8.94	50	1.2328	27.38	84	1.4427	44.49
17	1.0700	9.49	51	1.2384	27.91	85	1.4495	44.96
18	1.0744	10.04	52	1.2439	28.43	86	1.4564	45.44
19	1.0788	10.59	53	1.2496	28.96	87	1.4633	45.91
20	1.0833	11.15	54	1.2552	29.48	88	1.4702	46.37
21	1.0878	11.70	55	1.2609	30.00	89	1.4772	46.84
22	1.0923	12.25	56	1.2667	30.53	90	1.4842	47.31
23	1.0968	12.80	57	1.2724	31.05	91	1.4913	47.77
24	1.1014	13.35	58	1.2782	31.56	92	1.4984	48.23
25	1.1060	13.90	59	1.2841	32.08	93	1.5055	48.69
26	1.1107	14.45	60	1.2900	32.60	94	1.5126	49.14
27	1.1154	15.00	61	1.2959	33.11	95	1.5198	49.59
28	1.1201	15.54	62	1.3019	33.63	96	1.5270	50.04
29	1.1248	16.19	63	1.3079	34.13	97	1.5343	50.49
30	1.1296	16.63	64	1.3139	34.64	98	1.5416	50.94
31	1.1345	17.19	65	1.3200	35.15	99	1.5489	51.39
32	1.1393	17.73	66	1.3261	35.66	100	1.5563	51.93
33	1.1442	18.28	67	1.3323	36.16			

The above table is from the determinations of Dr. F. Plato, and has been adopted as standard by the United States Bureau of Standards.

Pumps for the Oil Industry

Petroleum Fields

The American Petroleum industry may be said to have had its inception from the date of the drilling of the first well in 1859. Prior to this date, petroleum had been obtained in small quantities from brine wells in Pennsylvania, and from the distillation of coal.

The Petroleum fields of importance in the United States may be listed as follows:

Appalachian
Lima—Indiana
Illinois
Mid-Continent
Central and Northern Texas
West Texas
Louisiana
Gulf Coast
Wyoming
California.

Chemical Properties of Petroleum

Petroleum is a mixture of chemical compounds of carbon and hydrogen called hydrocarbons with small amounts of sulphur, nitrogen and oxygen. These last three usually exist as derivatives of the hydrocarbons, and are regarded as impurities. Hydrogen sulphide, water and earthy matter are often present in addition.

The elements carbon and hydrogen of which all hydrocarbons are composed, possess widely different properties. Carbon which is one of the most widely distributed elements is the principal component of all organic compounds. Hydrogen is a colorless, odorless, inflamable gas and is the lightest substance known.

Physical Properties of Petroleum

Specific Gravity

Petroleum is lighter than water. The specific gravity is influenced by physical factors and by the chemical composition of the crude oil. American crude petroleum varies in specific gravity from .7684 to .9960. Russian petroleum from .854 to .889 and Mexican oil from .975 to .992. In practical operation in the Petroleum Industry, the specific gravity is generally expressed in terms of the Baumé scale which bears no direct relation to the specific gravity. The conversion of the Baumé scale into specific gravity is as follows:

$$\text{Degrees Baumé} = \frac{140}{\text{Sp. Gr. } 60°/60° \text{ Fah.}} - 130$$

$$\text{Sp. Gr. } 60°/60° \text{ Fah.} = \frac{140}{130 + \text{deg. Bé}}$$

Viscosity

The viscosity or measure of the tendency to flow is an important factor with lubricating oils, especially when it comes to handling them with pumps. It is usually determined as the time necessary for a definite volume of oil at a definite temperature to flow through a small opening or orifice. This work is carried out in an instrument known as a viscosimeter of which there are several standard makes in use at the present time. All of them utilize the same general principle. Oil is heated in a metallic cup, surrounded by an oil bath. The temperature of the oil in the cup and that of the oil bath are carefully controlled; and when the desired temperature has been reached, a small orifice in the bottom of the cup is opened, and the oil is allowed to flow into a flask of known capacity. The time necessary for the flask to fill is taken as the measure of the viscosity of the oil.

In the United States, the Saybolt standard uniform viscosimeter is generally used. The Redwood viscosimeter is used in Great Britain and in Germany, and other countries the Engler viscosimeter is used.

The time in seconds for the delivery of 60 cu. in. of oil is the Saybolt viscosity of the oil at the temperature of the test which generally is 100° Fah., 150° Fah. or 210° Fah.

The viscosity of fuel oils is determined by the Saybolt-Furol Viscosimeter at temperatures of 70° Fah. 104° Fah. and 122° Fah.

The viscosity of oil decreases and its fluidity increases when the temperature is raised. This is important as it has everything to do with the successful handling of it with pumps.

Refining

There are so many grades of crude and the method of refining them so varied that only an outline of the usual practice will be given here.

The process of treating crude petroleum is always one of fractional distillation with subsequent chemical treatment, filtering, re-distillation and compounding of each of the fractions. Often only one treatment is necessary to produce the desired result; often two or more steps are taken.

There are three major classifications of petroleum,

Paraffin Base.
Asphalt Base.
Mixed Base.

The paraffin base crude contains very little asphalt. Some of the well known paraffin base crudes are those from Pennsylvania, West Virginia, North Louisiana, Ranger, Oklahoma and Kansas.

The paraffin base crudes are most valuable. This is due not alone to the valuable lubricating oils and cylinder stocks contained therein, but also to the fact that such oils show a high yield of gasoline.

The asphalt base crudes leave as a residue a heavy pitch or asphalt. California, certain Texas and the heavy Mexican oils are of this class.

A mixed base petroleum is one in which both paraffin and asphalt are found. Illinois and some of the light Mexican oils are of this class.

The three classes of crudes are handled somewhat similarly, but the yields of the products are quite different. Paraffin base oils are usually run with every effort to avoid cracking or decomposition by heat. They yield the bulk of our high grade lubricants. Asphalt base crudes on the other hand, will yield greater amounts of like products when cracked. These oils are generally "topped" for the lighter fractions. Mixed base oils are usually refined by combination of the two processes, cracking and non-cracking.

The two methods employed in refining petroleum are straight distillation which separates the compounds of different boiling points from each other but does not break them up, and cracking which decomposes or breaks up the oil by heating it to a high temperature.

The demand for gasoline has long since exceeded the available supply contained in the crude oil, and a very important part of the refining practice today consists in augmenting the natural supply of light hydrocarbons by processes of decomposing the heavier oils by heat treatment and thereby obtaining increased quantities of motor fuel.

Straight Distillation

The crude is first distilled by direct firing with the aid of open steam sprays in the still. As the oil is heated in the still, those portions of low boiling points vaporize and are led through condensing coils to a so-called "look box." Here the stream is under observation of the still man who, by means of samples, keeps in touch with the operation. As the temperature of the oil in the still rises, fractions of high boiling points and higher gravity are vaporized and condensed, and flow through the look box. The look box is connected to a manifold leading to the receiving tanks, one or more tanks being provided for each "cut" or product desired from the distillation.

The cuts or separations resulting from the first distillation of a paraffin base crude will run about as follows:

Crude Naphtha start	54° Baumé
Kerosene Distillate	54°–50° Baumé
Crude Kerosene	50°–38° Baumé
Gas Oil	38°–35° Baumé
Wax Distillate	35° Baumé

The crude naphtha or first cut is the base of commercial naphthas and gasoline. The distillate will contain hydrocarbons of higher boiling points than are allowable in gasoline. Further distillation is therefore required, and this is carried out in steam stills with dephlegmating towers so the lighter constituents of the crude naphtha can escape. In this distillate 75 to 80% of the charge will pass over as commercial grades of gasoline. The residual which is naphtha bottoms is further distilled with kerosene stocks.

The kerosene distillate, like the naphtha cut, contains products which are more homogeneous. The distillate is treated in steam stills as before and run down to the desired flash point. The distillate which is carried over is put into the crude naphtha cut from the crude oil distillation for further handling.

The crude kerosene from the still and naphtha bottoms are then treated and represent various grades of kerosene oils.

Gas oil is usually marketed as obtained from the crude fire still. It is sometimes blended with heavier oils for producing certain grades of fuel oil. It is also used in the cracking processes. The wax distillate which is taken from the crude still contains many compounds of the solid paraffin series and also valuable oils from which high-grade lubricants are made. It is again subjected to distillation which changes the character of the wax to a crystalline form which enables it to be extracted. The resulting cracked distillate is chilled by refrigerating machines and then pumped through a filter press to remove the wax. The pressed oil is fire-stilled with bottom steam, the residual in the still resulting in a lubricating oil stock. The overhead distillates can then be fractionated, and when treated become the various grades of automobile, air compressor and engine oils. The wax taken from the distillate is sweated and purified, and becomes the commercial paraffin wax.

Cracking

The decomposition of petroleum with the consequent breaking up of the molecules and the production of both lighter and heavier hydrocarbons is called "cracking." In the oil industry, cracking processes are those designed to utilize the above decomposition for the conversion of heavier oils into the more valuable gasolines and naphthas. All petroleum hydrocarbons have a characteristic temperature above which the cracking reaction takes place. This temperature varies for the different cuts from the given crude and for similar cuts from different crude oils. For the gas oil and fuel oil distillates now in use as stock for cracking operation, the temperature necessary to cause the re-action is usually between 550° Fah. and 600° Fah.

The great bulk of the stocks which are cracked for yields of gasoline is composed of gas oils, and the stocks distilled from crude oil between that cut and coke. There are two essential factors in the cracking operation: temperature and the time

through which the molecule is subjected to that temperature. Pressure is corollary to the temperature. The simpler hydrocarbons require higher temperatures and longer time than the more complex ones of greater specific gravity. Such stocks as kerosene distillate are therefore seldom cracked on a commercial scale.

The systems in commercial use may be broadly classified as cracking in the vapor phase and cracking in the liquid phase.

Cracking in the vapor phase consists in vaporizing the oil and then producing the re-action by the addition of heat, or of heat and pressure together. Temperatures ordinarily used are from 1000° Fah. up, although it is possible to operate vapor phase cracking at as low a temperature as 750° Fah. Among the various processes which have been experimented upon in vapor phase are the Rittman, Gyro, Greenstreet and Ellis.

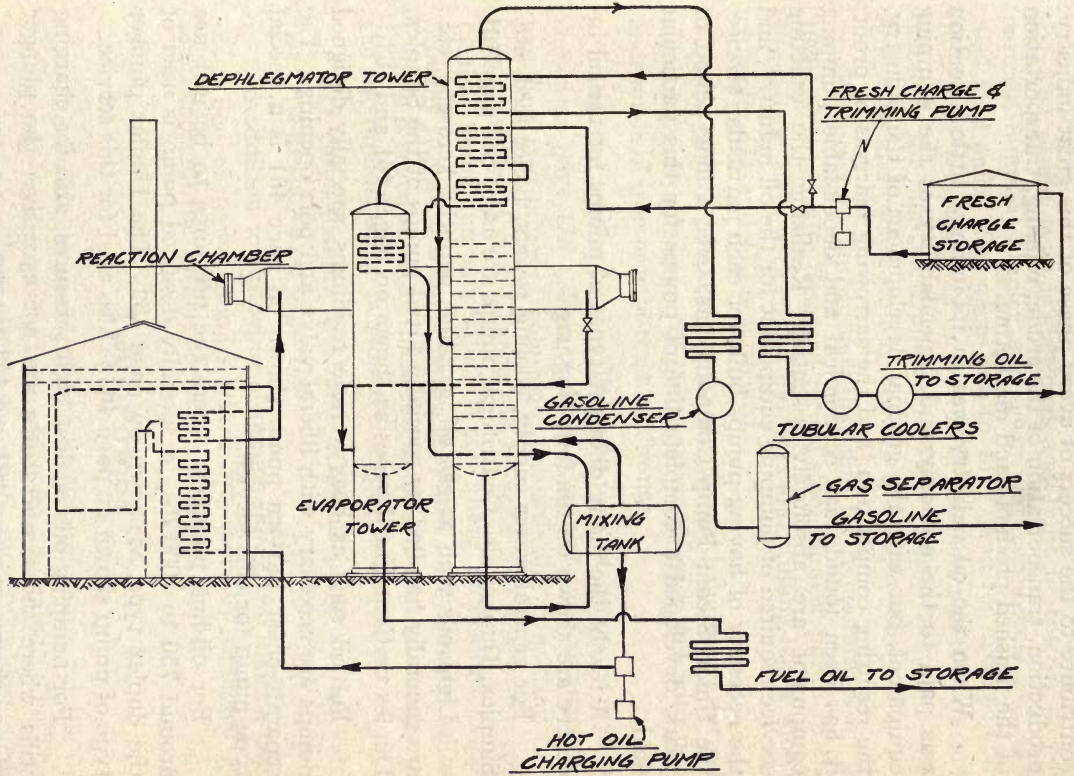
The majority of commercial processes in general use are on the liquid phase principle where the heat is applied to the liquid under very high pressure sufficient to keep it liquid. The processes in use are the Tube and Tank, Cross, Holmes-Manley, Dubbs, Fleming, Isom, Coast and Emerson. There are also many other processes but no attempt will be made here to describe them.

Casing Head Gasoline

Gasoline is also obtained from casing head gas which flows from oil wells between the tubing and the casing. There are two methods of handling this gas in the production of gasoline: by the compression method and by the absorption method.

When the gas is rich, it is compressed by a two-stage compressor at a pressure from 50 to 300 lbs. and flows through a series of water-cooled condenser coils in which the casing head gasoline separates and is collected in tanks, while the non-condensable vapors are piped away.

The absorption process is used where the gas yields less than 1½ gallons of gasoline per 1000 cubic feet. This process involves passing casing head gas under pressure through a heavy oil in which the heavier constituents of the gas are dissolved and retained by the heavy oil. By steam distillation of the heavy oil, the casing head gasoline is recovered.



Flow Diagram. For 2,000 Barrels per Day Hot-Oil Cross-Cracking Unit.

Pumps for the Oil Industry

In probably no other enterprise in the universe do pumps play such an important role as in the oil industry. Pumps contribute to the success of every phase of this great industry: production, transportation of the crude, refining, transportation of the finished products and marketing.

When the oil industry was in its infancy, little attention was paid to the economy and dependability of the pumping equipment, but with the gradual development of the industry, the resulting competition, and the narrowing margins of profit, it has been found that economical and dependable pumping equipment are absolutely essential. Realizing the exacting service required of pumping equipment for the oil industry, and the necessity of having pumps built for each particular service, the Union Steam Pump Company has, over a long period of years, developed a complete line of pumps for this particular field.

A few of the many different types of pumps for the oil industry are shown on the following pages.

For general water supply, cooling towers, boiler feed, etc., Simplex, Duplex, also Centrifugal pumps are used. See page 461.

Light oils such as gas oil, gasoline, naphtha kerosene, etc., are generally handled by specially fitted pumps of the Simplex, Duplex or Centrifugal types as shown on pages 461 and 464.

For oil such as heavy crude residiums, specially designed pumps are necessary with liberal valve and port openings. The separate-chest pattern pump shown on page 461 is a special type pump for this service.

For oil-line pumping, where the pressures to be met are around 750 to 1000 lbs., a specially designed pump is used of the duplex type, as shown on page 462. The Duplex plunger type of pump shown on page 463 is also used for this service.

For pumping oils against high pressures, also for charging stills where the temperatures are not excessive, the plunger pumps shown on pages 462 and 463 are used.

When it comes to the question of handling hot oils, the type of pump to use depends not only on the temperature but the pressure, the gravity of the oil, and the suction conditions. Every condition is different, so no definite statement can be

made as to the exact type to use until a careful study is made of the conditions.

In general, pumps for hot-oil service may be classed as piston and plunger types.

The piston type is designed and built in the valve pot pattern, Simplex, Duplex and Twin, with semi-steel and cast steel fluid ends. See pages 465 to 467. And for extreme conditions for temperatures up to 1000°, the piston pattern is furnished in the Simplex, Duplex and Twin types with forged steel fluid ends and cooled stuffing-boxes. See pages 468 to 470.

For pressures up to 3500 lbs., and for temperatures up to 1000°, the specially designed forged-steel plunger pump in the Simplex or Twin with cooled plungers and stuffing-boxes shown on pages 471 and 472 is furnished.

For handling heavy viscous oils like acid sludge, we furnish a specially designed type of pump shown on page 474 with fluid ends made of iron or acid-resisting bronze.

For fire protection, the Foamite type of pump is used as shown on page 473. This pump is a special type arranged for handling two different solutions; one solution consisting of a viscous material and bicarbonate of soda and the other containing aluminum sulphate. When these two solutions are brought together in equal parts, they will form a foam, whose volume is seven to eight times the combined original volumes of the two liquids, which is utilized in extinguishing fires.

For cargo loading, we build a special pump of the valve pot type as shown on page 474.

In addition to the types of pumps shown, we make a complete line of power pumps which are specially designed for practically all the various services mentioned above.

Twin Pumps

The Twin pumps referred to herein and shown on the following pages are a special type which we originated. The Twin pump consists of two Burnham single pumps taking steam through a patented synchronizing valve, which device keeps the two pumps in step, one pump reversing when the other is at about mid-stroke. The synchronizing valve is functioned by the auxiliary pistons carrying the main steam valve of the two

companion pumps, and this mechanism is entirely independent of the valve gear which controls the stroke of the two pumps.

The Burnham twin mechanism is a simple device by means of which it is possible for us to furnish a pumping unit which is particularly adapted to handling oils. A few of the many advantages of this pump are:

A pump with a very uniform discharge which is essential.

Flexibility, which is of paramount importance for a pump for this service.

The pump may be operated as a twin or in case of repacking, repairs, etc., either side of the unit may be shut down, the other continuing to run at practically double the speed.

By utilizing the Burnham Simplex steam end, it is possible to obtain a high degree of economy which is characteristic of that pump.

Since each pump is operated by its own valve gear, there is no possibility of short-stroking as the pump must take its full stroke before it can reverse. This feature is particularly desirable for handling oils, also if the pump should be used as a meter.

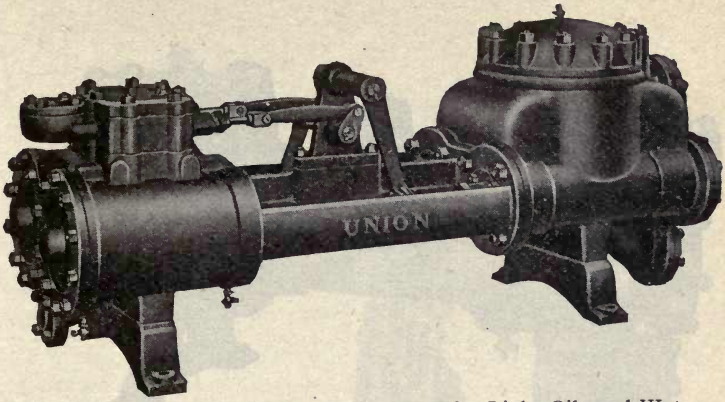


Fig. 103a. General Service Duplex Pump for Light Oils and Water.

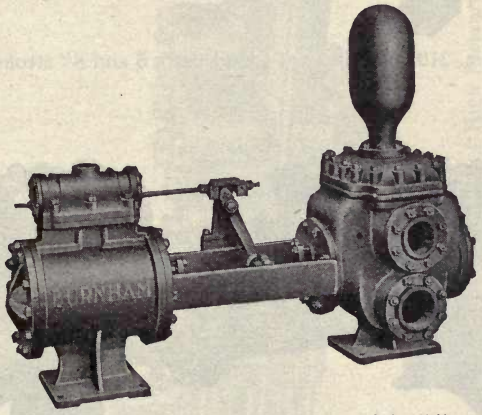


Fig. 95. General Service Simplex Pump for Light Oils and Water.

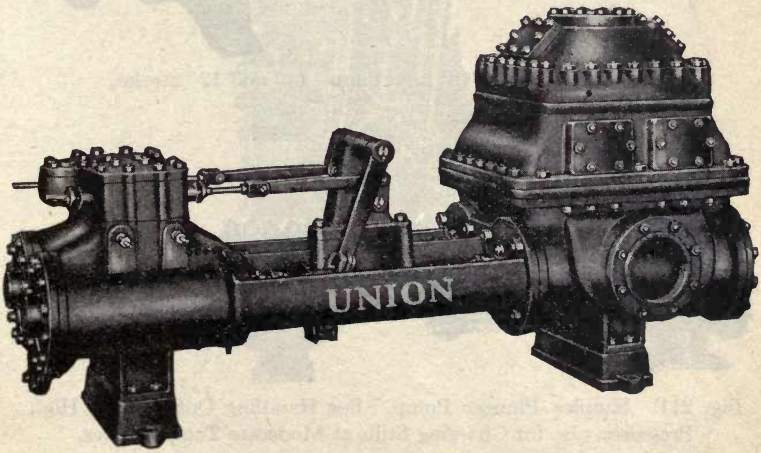


Fig. 205. Separate-Chest Pattern Duplex Pump for Handling Heavy Oil.

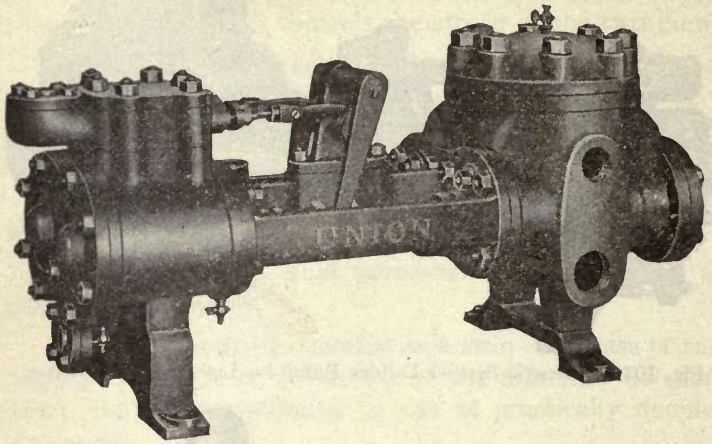


Fig. 210. Duplex Oil Line Pump, 6 and 8″ stroke.

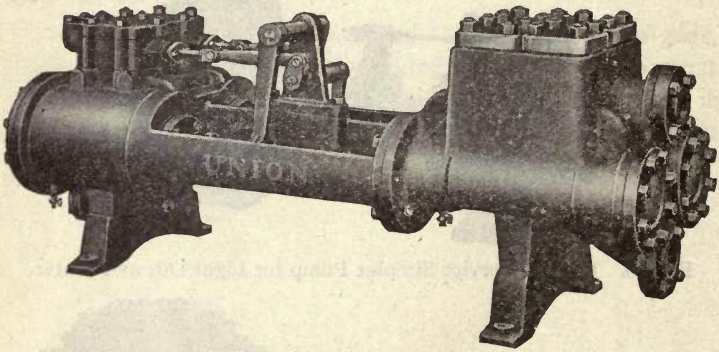


Fig. 181. Duplex Oil Line Pump, 10″ and 12″ stroke.

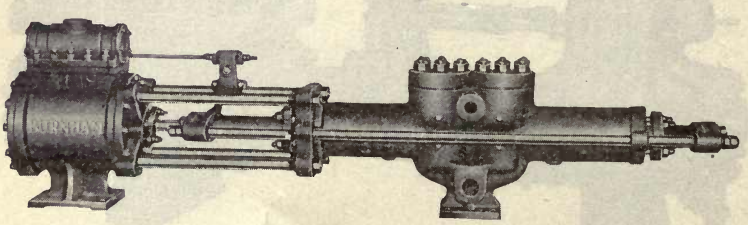


Fig. 211. Simplex Plunger Pump. For Handling Oils Against High Pressures, also for Charging Stills at Moderate Temperatures.

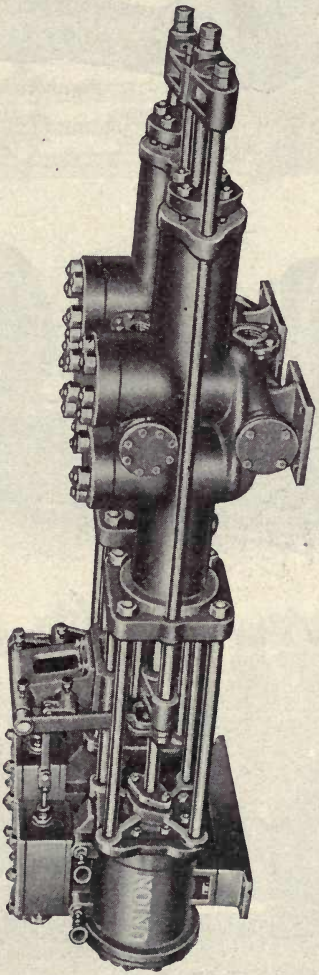


Fig. 212.

Duplex Plunger Pump for Oil Line Pumping, also for High Pressure and Charging Stills at Moderate Temperatures.

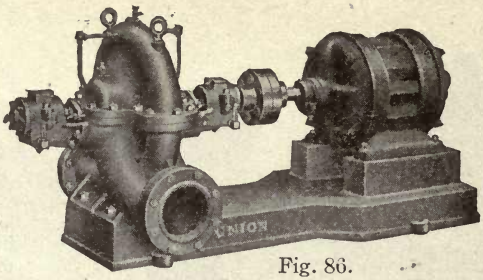


Fig. 86.

Motor-Driven Pump. For Circulating Light Oils, Water, etc.

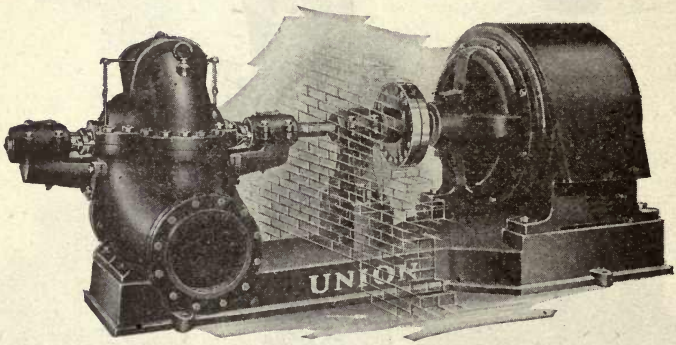


Fig. 213.

Motor-Driven Gasoline Pump.

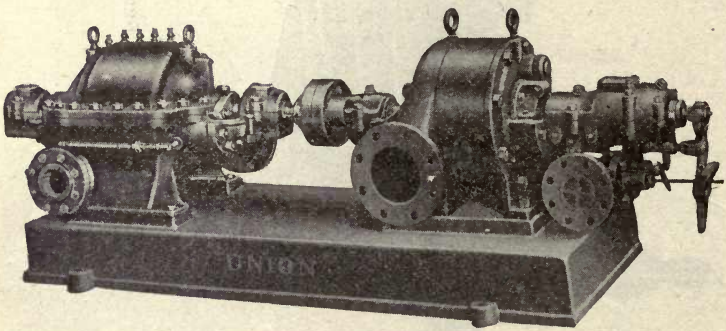


Fig. 214.

Multistage Steam Turbine-Driven Centrifugal Pump. For High Pressure Oil or Water, also for Boiler Feeding.

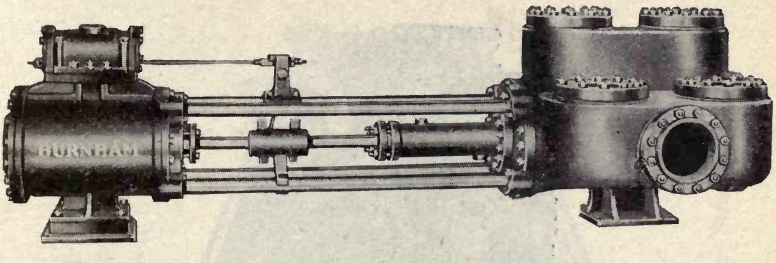


Fig. 215.

Burnham Valve-Pot Pattern Simplex Hot Oil Pump.

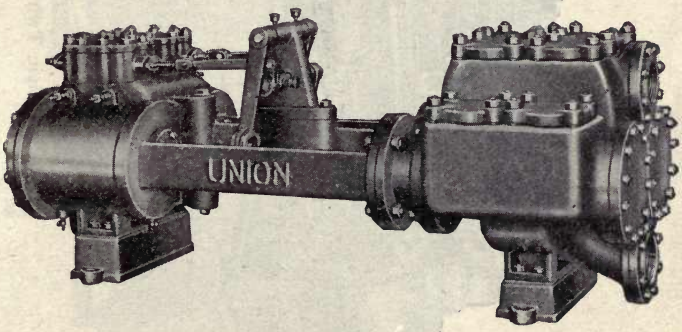


Fig. 216.

Duplex Valve-Pot Pattern Hot Oil Pump.

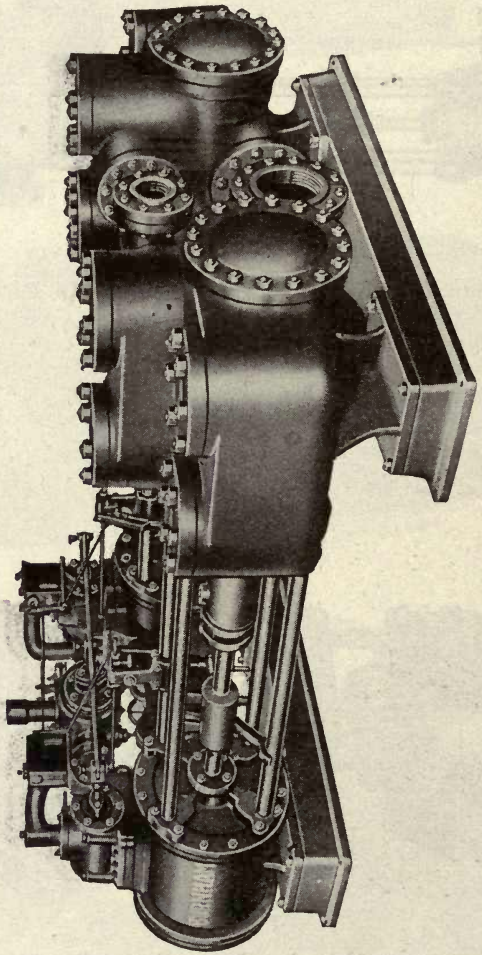


Fig. 217.
Burnham Twin Valve-Pot Pattern Hot Oil Pump.

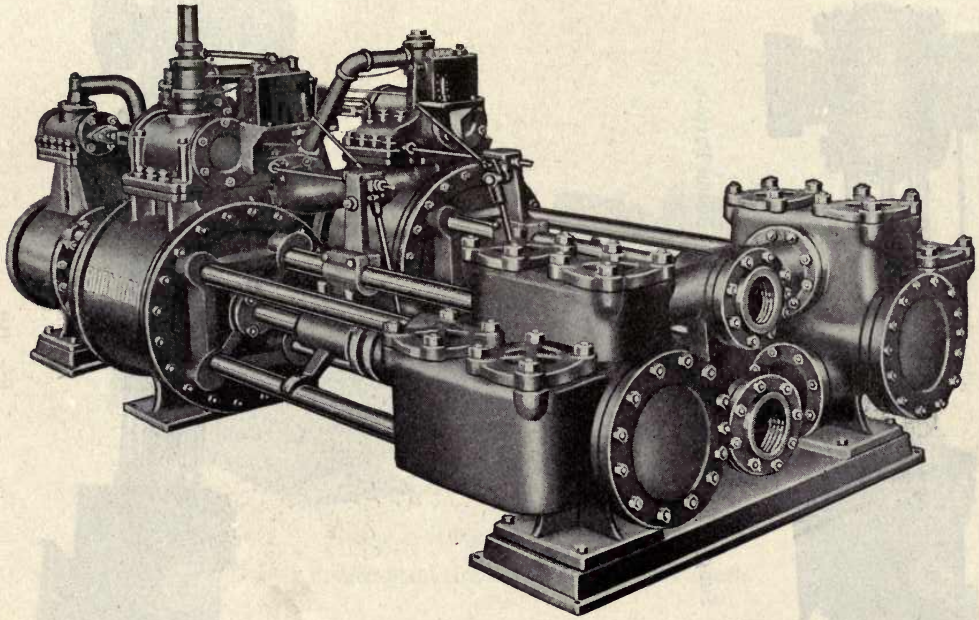


Fig. 230.

Burnham Twin Valve-Pot Pattern Hot Oil Pump with Compound Steam End.

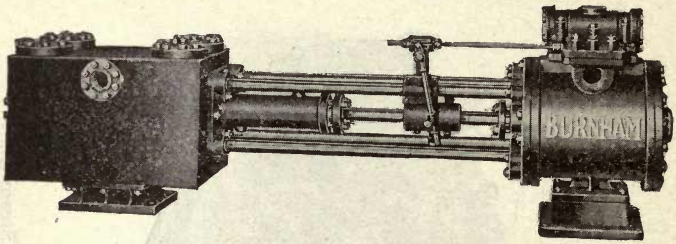


Fig. 218.

Simplex Forged-Steel Hot Oil Pump, Piston Pattern.

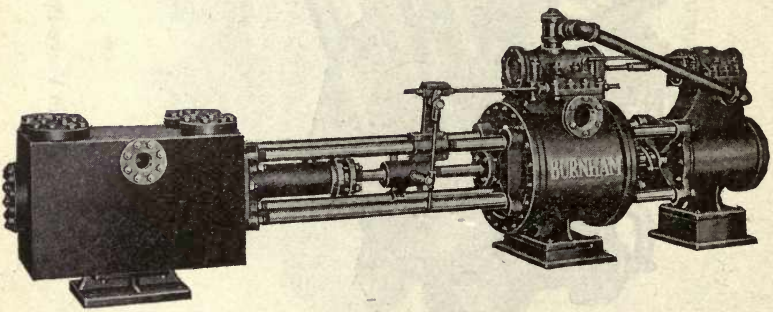


Fig. 219.

Simplex Forged-Steel Hot Oil Pump, Piston-Pattern with Compound Steam End.

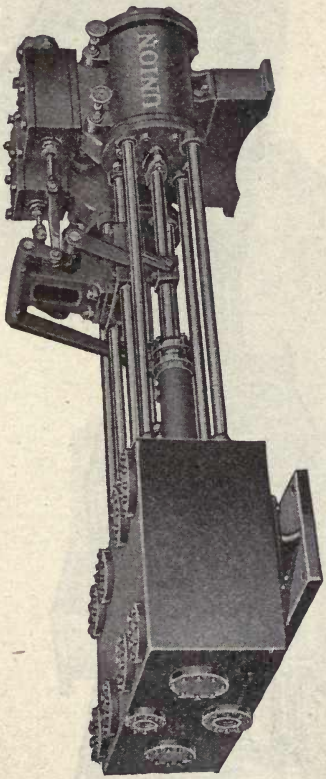


Fig. 220.

Duplex Forged-Steel Hot Oil Pump, Piston Pattern.

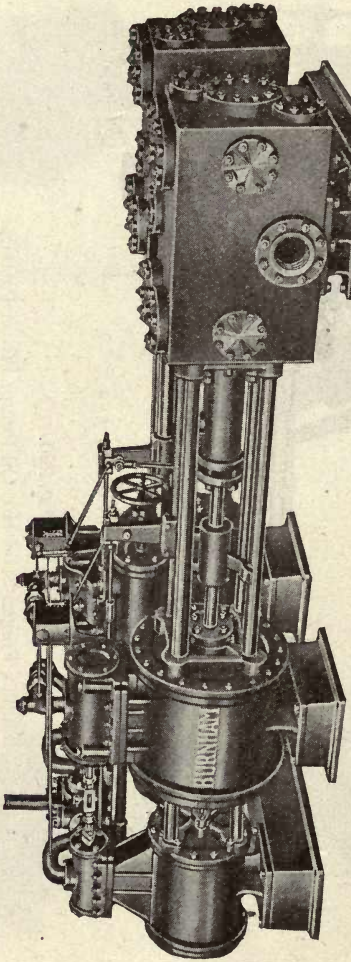


Fig. 221.

Twin Forged-Steel Hot Oil Pump, Piston Pattern with Compound Steam End.

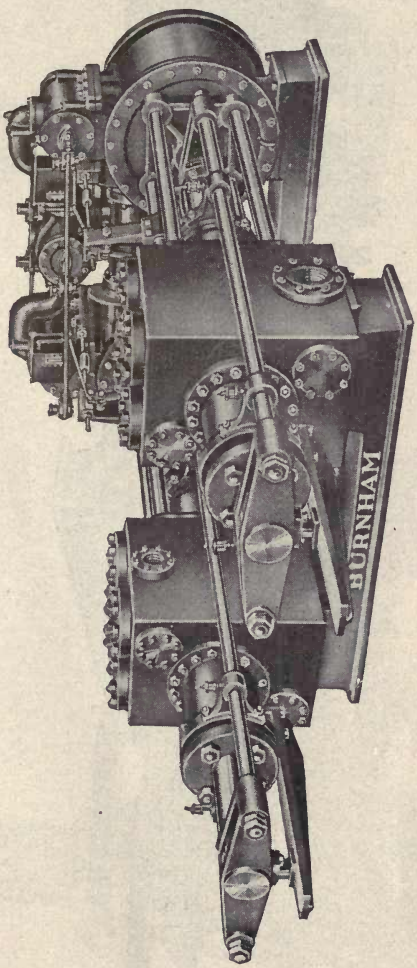


Fig. 222.
Twin Forged-Steel Hot Oil Pump, Plunger Pattern.

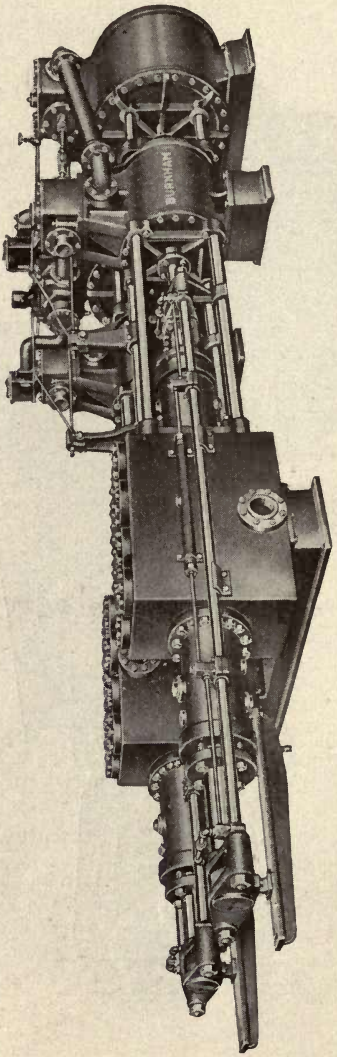


Fig. 223.

Twin Forged-Steel Hot Oil Pump, Plunger Pattern with Compound Steam End

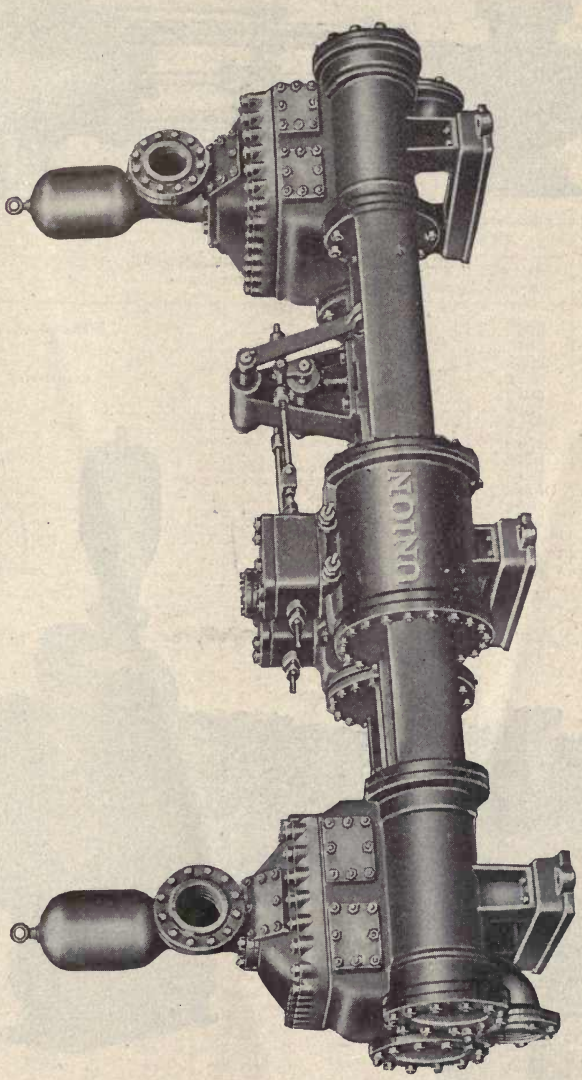


Fig. 224.
Duplex Foamite Pump.

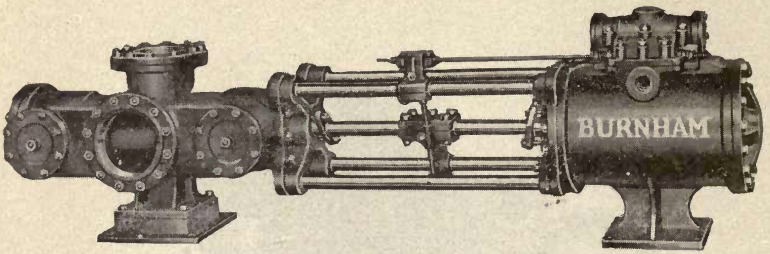


Fig. 225.

Pump for Heavy Oils, Sludge, etc.

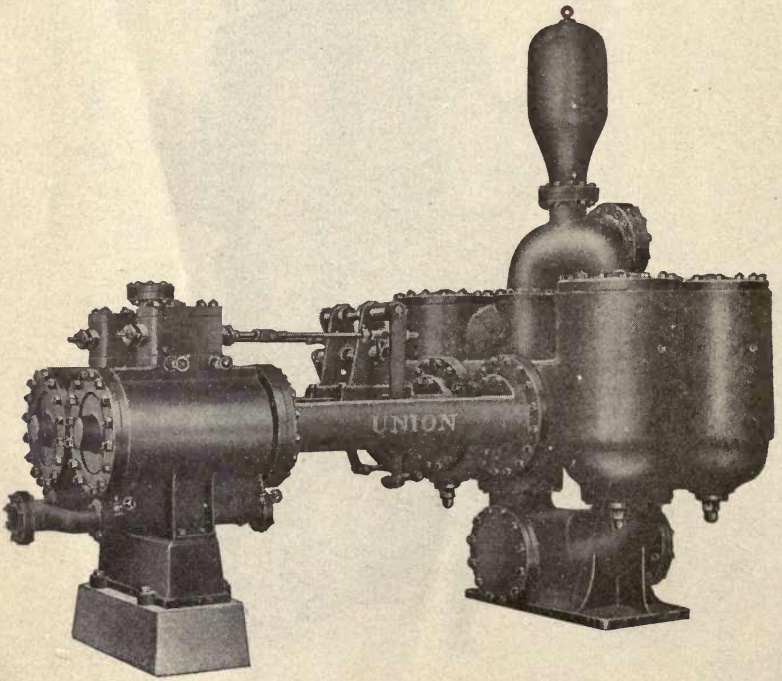


Fig. 182.

Cargo Loading Pump.

Useful Information---Oil

1 Barrel equals 42 U. S. gals.
1 Barrel per hour equals .7 U. S. gals. per minute.
Barrels per hour x .7 equals gals. per minute.
Gals. per minute divided by .7 equals barrels per hour.
Barrels per hour x 24 equals barrels per day.
1 Barrel per day equals .0292 gals. per minute.
Barrels per day x .0292 equal gals. per minute.
Gals. per minute divided by .0292 equals barrels per day.
Number of barrels in pipe one mile long equals diameter of pipe in inches squared x 5⅛.
Velocity in feet per second equals .0119 x barrels per day divided by diameter of pipe in inches squared, or Velocity equals .2856 x barrels per hour divided by the diameter of pipe in inches squared, or Velocity equals .408 x gals. per minute divided by the diameter of pipe in inches squared.
Net Horse Power equals the theoretical horse power necessary to do the work.
Net Horse Power equals barrels per day x Pressure x .000017.
Net Horse Power equals barrels per hour x Pressure x .000408.
Net Horse Power equals gals. per minute x Pressure x .000583.

CHARACTERISTICS OF TESTED OILS

OILS TESTED	GRAVITY		Pounds per Gallon
	Degrees Beaume	Specific Gravity	
California, Bakersfield	16.0	.9595	7.994
Louisiana, Jennings	24.0	.9105	7.585
Ohio, Lima	37.0	.8395	6.994
Oklahoma Residuum	28.0	.8870	7.390
" "	24.0	.9105	7.585
" Crude, (G. P.)	32.2	.8631	7.198
" "	36.2	.8423	7.023
" "	35.4	.8464	7.055
Pennsylvania	43.7	.8059	6.722
"	38.2	.8323	6.944
Russian, Baku	29.0	.8815	7.344
Texas, Beaumont	22.0	.9220	7.681
" , Sour Lake	20.0	.9340	7.781
" , Residuum	18.0	.9465	7.885
" Gas Oil	27.9	.8866	7.394
West Virginia	40.0	.8250	6.873

FRICTION LOSS IN OIL PIPE LINES

The friction loss in oil pipe lines may be found by the following formula:

$$F = \frac{C \times B^2}{10 \times D^5}$$

In which

F = Friction in pounds per square inch per mile (5280 feet.)
C = A Constant from the table below which depends on the character of the oil.
B = Barrels of oil per hour.
D = Diameter of pipe in inches.

TABLES COMPUTED FOR 38° BEAUME

When "F" is given and "B" computed, add 1% to "B" for every 3° above and subtract 1% from "B" for every 3° below: When "B" is given and "F" computed, subtract 2% from "F" for every 3° above and add 2% for every 3° below—or interpolate for 9.00

CONSTANTS FOR DIFFERENT OILS

Degrees Beaume	Const.	Degrees Beaume	Const.	Degrees Beaume	Const.	Degrees Beaume	Const.
65	7.38	53	8.10	41	8.82	29	9.54
62	7.56	50	8.28	38	9.00	26	9.72
59	7.74	47	8.46	35	9.18	23	9.90
56	7.92	44	8.64	32	9.36	20	10.08

Using 9.00 as a CONSTANT for 38° oil, subtract .06 for each degree above 38°, and add .06 for each degree below 38°.

For every 10 degrees above 60 degrees Fahrenheit, subtract 1° from the Beaume reading, and for every 10 degrees below 60 degrees Fahrenheit add 1° to the Beaume reading;

For Example:—

Raise 38° Bme oil from 60° F to 80°F; then 80—60 = 20 and 20 ÷ 10 = 2: and 38—2 = 36 or 36° Bme the new Beaume reading for 36° Bme Oil heated from 60°F to 80°F.

Characteristics of American Petroleum Oils

Deg. Baume	U. S. Bureau of Standards		A. P. I.		Change in Sp. Grv. per Deg. F.	Deg. Baume	U. S. Bureau of Standards		A. P. I.		Change in Sp. Grv. per Deg. F.
	Specific Gravity	Pounds per Gallon	Specific Gravity	Pounds per Gallon			Specific Gravity	Pounds per Gallon	Specific Gravity	Pounds per Gallon	
10.0	1.0000	8.325	1.0000	8.331	.00035	30.0	0.8750	7.286	0.8762	7.300	.00036
10.5	.9964	8.299	.9965	8.302	.00035	30.5	.8723	7.264	.8735	7.277	.00037
11.0	.9929	8.269	.9930	8.273	.00035	31.0	.8696	7.241	.8708	7.255	.00037
11.5	.9894	8.240	.9895	8.244	.00035	31.5	.8669	7.218	.8681	7.232	.00037
12.0	.9859	8.211	.9861	8.215	.00035	32.0	.8642	7.196	.8654	7.210	.00037
12.5	.9825	8.182	.9826	8.186	.00035	32.5	.8615	7.173	.8628	7.188	.00037
13.0	.9790	8.153	.9792	8.158	.00035	33.0	.8589	7.152	.8602	7.166	.00037
13.5	.9756	8.125	.9759	8.130	.00035	33.5	.8563	7.130	.8576	7.145	.00037
14.0	.9722	8.096	.9725	8.102	.00035	34.0	.8537	7.108	.8550	7.123	.00037
14.5	.9688	8.069	.9692	8.074	.00035	34.5	.8511	7.087	.8524	7.104	.00037
15.0	.9655	8.041	.9659	8.047	.00035	35.0	.8485	7.065	.8498	7.080	.00037
15.5	.9622	8.013	.9626	8.019	.00035	35.5	.8459	7.044	.8473	7.059	.00037
16.0	.9589	7.986	.9593	7.992	.00035	36.0	.8434	7.022	.8448	7.038	.00037
16.5	.9556	7.959	.9561	7.965	.00035	36.5	.8408	7.001	.8423	7.017	.00037
17.0	.9521	7.931	.9529	7.939	.00035	37.0	.8383	6.980	.8398	6.996	.00037
17.5	.9492	7.904	.9497	7.912	.00035	37.5	.8358	6.960	.8373	6.976	.00037
18.0	.9459	7.877	.9465	7.885	.00035	38.0	.8333	6.939	.8348	6.955	.00038
18.5	.9428	7.851	.9433	7.859	.00035	38.5	.8309	6.918	.8324	6.935	.00038
19.0	.9396	7.825	.9402	7.833	.00035	39.0	.8284	6.898	.8299	6.914	.00038
19.5	.9365	7.799	.9371	7.807	.00035	39.5	.8260	6.877	.8275	6.894	.00038
20.0	.9333	7.772	.9340	7.781	.00036	40.0	.8235	6.857	.8251	6.874	.00038
20.5	.9302	7.747	.9309	7.755	.00036	40.5	.8211	6.837	.8227	6.854	.00039
21.0	.9272	7.721	.9279	7.730	.00036	41.0	.8187	6.817	.8203	6.834	.00039
21.5	.9241	7.696	.9248	7.705	.00036	41.5	.8163	6.797	.8179	6.814	.00039
22.0	.9211	7.670	.9218	7.680	.00036	42.0	.8140	6.777	.8156	6.795	.00039
22.5	.9180	7.645	.9188	7.655	.00036	42.5	.8116	6.758	.8132	6.775	.00039
23.0	.9150	7.620	.9159	7.630	.00036	43.0	.8092	6.738	.8109	6.756	.00039
23.5	.9121	7.595	.9129	7.605	.00036	43.5	.8069	6.718	.8086	6.736	.00039
24.0	.9091	7.570	.9100	7.581	.00036	44.0	.8046	6.699	.8063	6.717	.00039
24.5	.9061	7.546	.9071	7.557	.00036	44.5	.8023	6.680	.8040	6.698	.00039
25.0	.9032	7.522	.9042	7.533	.00036	45.0	.8000	6.661	.8017	6.679	.00039
25.5	.9003	7.497	.9013	7.509	.00036	45.5	.7977	6.642	.7994	6.660	.00039
26.0	.8974	7.473	.8984	7.485	.00036	46.0	.7955	6.623	.7972	6.641	.00039
26.5	.8946	7.449	.8956	7.461	.00036	46.5	.7932	6.604	.7949	6.623	.00040
27.0	.8917	7.425	.8927	7.437	.00036	47.0	.7910	6.586	.7927	6.604	.00040
27.5	.8889	7.402	.8899	7.414	.00036	47.5	.7887	6.567	.7905	6.586	.00040
28.0	.8861	7.378	.8871	7.390	.00036	48.0	.7865	6.548	.7883	6.567	.00040
28.5	.8833	7.355	.8844	7.378	.00036	48.5	.7843	6.530	.7861	6.549	.00040
29.0	.8805	7.332	.8816	7.345	.00036	49.0	.7821	6.511	.7839	6.531	.00041
29.5	.8777	7.309	.8789	7.322	.00036	49.5	.7799	6.494	.7818	6.513	.00041

Friction of Oil — 38° Beaumé

Pounds Per Square Inch in Pipes 1 Mile Long

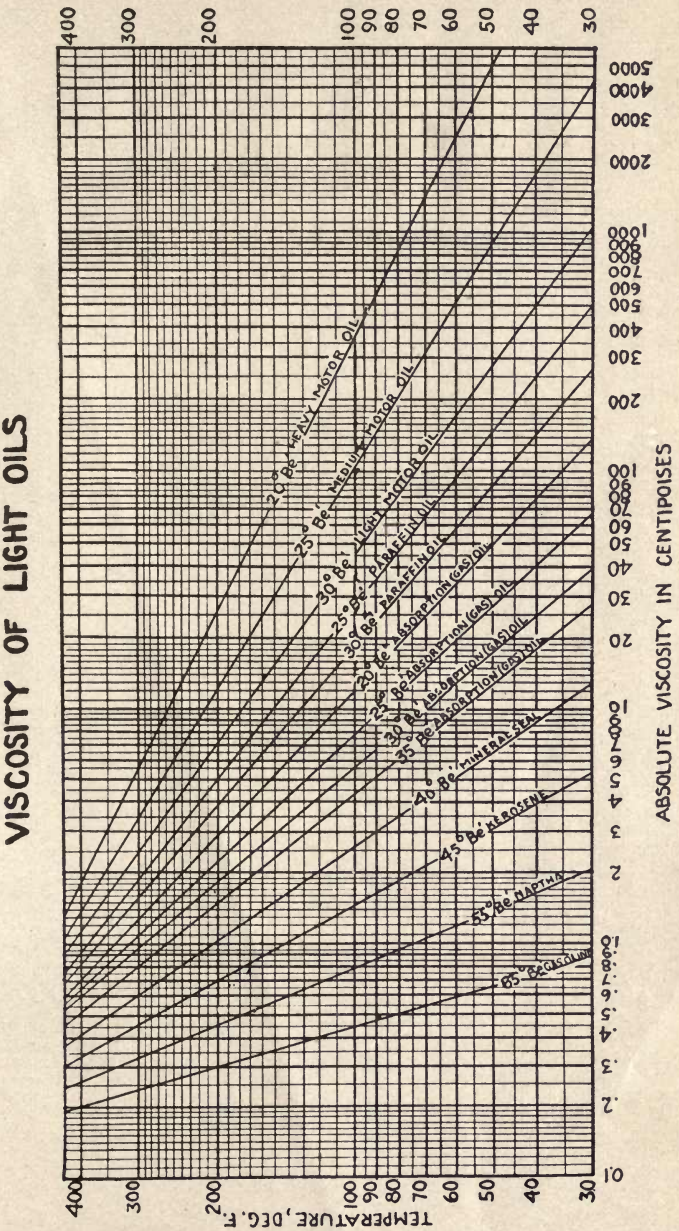
Barrels per Hour	DIAMETER OF PIPE						
	2″	3″	4″	5″	6″	8″	10″
10	2.8	.37	.088	.0288	.0116	.00274	.00091
15	6.2	.83	.198	.063	.026	.0064	.00203
20	11.2	1.48	.352	.115	.0462	.011	.0036
25	17.6	2.3	.55	.18	.0725	.0172	.0056
30	25.3	3.32	.792	.26	.104	.0247	.0081
35	34.5	4.52	1.08	.353	.142	.0337	.0112
40	45.0	5.9	1.43	.46	.185	.044	.0144
45	57.0	7.49	1.78	.58	.235	.0556	.0181
50	70.	9.24	2.2	.72	.29	.0686	.0225
55	85.	11.2	2.26	.87	.35	.084	.0273
60	101.	13.3	3.16	1.04	.416	.099	.0325
65	118.4	15.6	3.72	1.22	.49	.116	.0381
70	138.	18.1	4.3	1.41	.568	.136	.0442
75	158.	20.8	4.95	1.62	.651	.155	.0508
80	180.	23.6	5.62	1.84	.74	.176	.0578
85	203.	26.7	6.35	2.08	.836	.198	.0652
90	227.	30.0	7.11	2.33	.939	.224	.073
95	253.	33.4	7.92	2.6	1.045	.248	.0814
100	280.	37.0	8.8	2.88	1.16	.275	.0902
125	440.	57.8	13.7	4.5	1.81	.43	.142
150	630.	83.0	19.8	6.5	2.6	.618	.203
175		113.0	27.0	8.8	3.54	.84	.275
200		148.0	35.2	11.5	4.63	1.1	.36
225		187.	42.5	14.6	5.85	1.39	.455
250		230.	55.0	18.0	7.22	1.72	.56
275		280.	66.5	21.7	8.75	2.08	.68
300		332.	79.2	26.0	10.4	2.47	.81
325		390.	93.0	30.4	12.2	2.9	.95
350		453.	10.2	35.3	14.2	3.37	1.12
375			124.	40.5	16.3	3.85	1.26
400			141.	46.0	18.5	4.4	1.44
425			159.	52.0	20.9	4.96	1.63
450			178.	58.2	23.5	5.56	1.83
475			198.	65.0	26.1	6.25	2.04
500			220.	72.0	28.9	6.88	2.25
525			242.	79.5	31.9	7.58	2.48
550			266.	37.0	35.0	8.3	2.73

Friction of Oil — 38° Beaumé

Pounds Per Square Inch in Pipes 1 Mile Long

(Continued)

Barrels per Hour	Diameter of Pipe				Barrels per Hour	Diameter of Pipe	
	5″	6″	8″	10″		8″	10″
550	87.0	35.0	8.3	2.72	3600	356	117
575	95.0	38.4	9.1	2.98	3800	397	130
600	104	41.7	9.9	3.24	4000	440	144
625	112	45.4	10.7	3.5	4200		158
650	122	49.0	11.6	3.8	4400		174
675	131	53.0	12.5	4.1	4600		192
700	141	56.9	13.5	4.42	4800		207
725	152	61.0	14.4	4.73	5000		225
750	162	65.3	15.5	5.06	5200		243
775	173	69.8	16.5	5.41	5400		262
600	184	74.3	17.6	5.76	5600		282
825	196	79.0	18.7	6.12	5800		303
850	208	83.9	19.8	6.5	6000		324
875	221	88.8	21.1	6.89	6200		346
900	233	94.0	22.3	7.3	6400		368
925	247	99.2	23.5	7.7	6600		392
950	260	104.5	24.8	8.12	6800		416
975	274	111.0	26.2	8.55	7000		441
1000	288	115.8	27.5	9.0			
1100	348	140.0	33.2	10.9			
1200	415	166.5	39.5	13.0			
1300	486	195.5	46.4	15.2			
1400		227.0	53.8	17.6			
1500		261.0	61.8	20.3			
1600		296.0	70.4	23.1			
1700		335.0	79.3	26.1			
1800		375.0	89.0	29.2			
1900		418.0	99.0	32.5			
2000		463.0	110.0	36.1			
2200			133.0	43.6			
2400			158.0	52.0			
2600			186.0	61.0			
2800			216.0	70.6			
3000			247.5	81.1			
3200			284.0	92.2			
3400			318.0	104.0			
3600			356.0	117.0			


VISCOSITY OF LIGHT OILS
TEMPERATURE, DEG. F.
ABSOLUTE VISCOSITY IN CENTIPOISES
400
300
200
100
90
80
70
60
50
40
30
5000
4000
3000
2000
1000
800
700
600
500
400
300
200
100
90
80
70
60
50
40
30
20
15
10
7
6
5
4
3
2
1.6
1.5
1.4
1.3
1.2
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
20° Be' HEAVY MOTOR OIL
25° Be' MEDIUM MOTOR OIL
30° Be' LIGHT MOTOR OIL
25° Be' PARAFFIN OIL
30° Be' PARAFFIN OIL
25° Be' ABSORPTION (GAS) OIL
30° Be' ABSORPTION (GAS) OIL
35° Be' ABSORPTION (GAS) OIL
40° Be' MINERAL SEAL
40° Be' KEROSENE
45° Be' KEROSENE
53° Be' NAPTHA
65° Be' GASOLINE

THIS BOOK IS DUE ON THE LAST DATE STAMPED BELOW

AN INITIAL FINE OF 25 CENTS
WILL BE ASSESSED FOR FAILURE TO RETURN THIS BOOK ON THE DATE DUE. THE PENALTY WILL INCREASE TO 50 CENTS ON THE FOURTH DAY AND TO $1.00 ON THE SEVENTH DAY OVERDUE.

SEP 21 1940
SEP 22 1940
OCT 9 1941
APR 27 1942 U
MAY 7 1943
NOV 11 1943
JAN 4 1946
2Apr'49BL
4 Apr'49AP
REC'D LD NOV 7'72

29Mar'50GE
JONES
MAR 29 1950
APR 29 1950
12 Oct'50CD
15Apr'55PW
APR 1 1955 LU
5Nov'63SB
REC'D LD
OCT 22'63 -11 AM
NOV 7 1972 60
1 AM 23

LD 21-100m-7/39(402s)

TB 52024

Paul Christiansen

M73420

TJ900
07
1921

THE UNIVERSITY OF CALIFORNIA LIBRARY